

Rolling Bearing Application

Liu Zejiu He Shiquan Liu Hui



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This book briefly introduces the types of rolling bearings and their application, identification codes, tolerance and rotating accuracy, general international technical terms, dynamic and static capacity and life calculation, elastic and permanent deformation calculation, support arrangement, fit selection, radial internal clearance selection, rolling bearing vibration and noise, sliding friction in rolling bearings and lubricant, lubrication method selection, bearing mounting and dismounting and so on.

This would be a book reference for all kinds of the technical engineers and academia person who want to know the basic knowledge of rolling bearings and who are going to select a proper rolling bearing or who carry on bearing applied design. Also this book can be a reference book for all the persons who use rolling bearing and for the teachers and students of universities and colleges.

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Preface

There have been three major changes or corrections in technical term of China bearing industry since 1970:

1. The change of load capacity and life calculation method of rolling bearings;

Before 1970, China rolling bearing industry used the load capacity and life calculation method of the former Soviet Union. That is the working capacity factor method of rolling bearings, Its basic equation is: $C = (Qnh)^{0.3}$. Where, C —working capacity factor of a rolling bearing; for a specific rolling bearing, it is a constant. Q —actual load on a rolling bearing (kN). n —rolling bearing rotation speed (r/min). h —rolling bearing life (h).

G. Lundberg and A. Palmgren published two famous articles in 1945 and 1952 respectively, and put forward a statistical probability method for rolling bearing capacity and life calculation. Its basic equation is: For ball bearing, $L = (C/P)^3$; For roller bearing, $L = (C/P)^{10/3}$. Where, C —rolling bearing basic dynamic rating; for a specific rolling bearing, it is a constant (kN). P —equivalent dynamic load (kN). L —rolling bearing life (10^6 r).

Since the publishing of the two famous articles, many rolling bearing companies have carried on a lot of studying, testing, and verifying work about this new rolling bearing theory. The International Standard Organization, ISO, published two recommended standards:

ISO/R281—1962 Basic dynamic load rating of rolling bearings;
ISO/R76—1962 static load rating of rolling bearings. Till 1970, most countries adopted the two ISO recommended standards. So China rolling bearing industry decided to adopt these two ISO standards in China, and stopped using the old rolling bearing calculation method. In 1975, the Rolling Bearing Sample Book was published by China Machine Press, in

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which for the first time the parameters of C , C_0 and rotation speed limit n for every rolling bearing were given out.

The change of rolling bearing life calculation method would affect many aspects. and all technical data about rolling bearings must be changed accordingly. Therefore, in 1978. we wrote the book *Rolling Bearing Load Rating and Life* and the book was published by China Machine Press, in 1982.

2. The identification code change of rolling bearings

Before 1990, China rolling bearing industry used the former Soviet Union's rolling bearing identification codes (designation) which was expressed by numbers. It was quite different from the common international rolling bearing identification code which was expressed by English letters and numbers. China government carried out the reform in 1978, and China rolling bearing export increased very quickly. But the difference rolling bearing identification codes seriously hurt Chinese rolling bearing export. Therefore, China rolling bearing industry decided to change the old identification code and adopted the commonly used international rolling bearing identification codes in 1990 and stipulated the new China National Standard GB/T272—1993: rolling bearing identification codes, and it took effect from the first of July, 1994. The identification code change of Chinese rolling bearings affected more comprehensive aspects. So, in 1994, we wrote the book *Rolling Bearing Application Handbook*, which was published by China Machine Press in 1996.

3. The technical standard revision of rolling bearings

The International Standard Organization (ISO) decided to restore Chinese membership in 1978. The China rolling bearing industry hold a meeting of ISO/TC4 (The Rolling Bearing Technical Committee, that is the fourth committee of ISO) in Hangzhou of China in April of 1986.

Since then, China rolling bearing technical standards have been revised synchronized with the international rolling bearing standards, Now most of the international rolling bearings standards have been adopted as the China National Standards equally or equivalently. So that, we wrote the book

Rolling Bearing Application Handbook (2ed) and the book was published by China Machine Press, in January of 2006.

Based on latest information and development of rolling bearing, recently we wrote the book *Rolling Bearing Application*. This book briefly introduces the types of rolling bearing and their application, identification codes, tolerance and rotating accuracy, common international technical terms, dynamic and static capacity and life calculation, elastic and permanent deformation calculation, support arrangement, fit selection, radial internal clearance selection, rolling bearing vibration and noise, sliding friction in rolling bearings, lubricant, lubrication method selection, bearing mounting and dismounting and so on.

The Chinese edition of this book was written by Liu Zejiu (Chapter 1, 2, 4, 6), He Shiquan (Chapter 3, 7) and Liu Hui (Chapter 5). The English edition of this book was translated by the above authors.

It is hard job to produce this book in English version. We are very thankful to China Machine Press. We also did appreciate any valuable comments from all of readers.

Liu Zejiu

List of Symbols

A	minimum load Constant; Constant
a	Semi-major axis of projected contact ellipse
a_1	Life adjustment factor for reliability
B	Bearing width
b	Semi-minor axis of projected contact ellipse
C	Dynamic load rating
C_0	Static load rating
C_a	Semi-major axis coefficient of projected contact ellipse
C_b	Semi-minor axis coefficient of projected contact ellipse
C_σ	Maximum contact stress coefficient
C_δ	Relative elastic approach coefficient
D	Outside diameter of rolling bearing
D_{pw}	Pitch diameter of rolling bearing
D_m	Average diameter of rolling bearing
D_w	Rolling element diameter
d	Innerring diameter of rolling bearing
E	Modulus of elasticity
F	Applied load on support system
F_A	Applied axial load on support system
F_a	Applied axial load on rolling bearing
F_r	Applied radial load on rolling bearing
F_t	Gear drive circumference force
F_s	Gear drive radial force
f	Curvature radius coefficient of raceway groove
i	Numbers of rolling element rows in rolling bearing
J_a	Axial integral of load distribution
J_r	Radial integral of load distribution
J_1	Mean rolling element load integral of rotating ring

J_2	Mean rolling element load integral of static ring
K	Elastic deformation constant
L	Rating life of rolling bearing
L_s	Fatigue life corresponding to survival probability
L_{10}	Basic rating life that 10% of a group of rolling bearings will endure
L_m	Median life that 50% of a group of rolling bearing will endure
L_{na}	Adjusted fatigue life of arbitrary condition
L_{we}	Effect roller length
M	Bending moment
$\frac{1}{m}$	Poisson's ratio
N	Drive power
n	Rotation speed of rolling bearing
n_i	Rotation speed of inner ring
n_e	Rotation speed of outer ring
n_c	Rotation speed of cage
n_{ci}	Cage rotation speed relative with inner ring
n_{ec}	Outer ring rotation speed relative with cage
n_w	Rolling element rotation speed around self-center axis, or rolling element self rotation speed
O_i	Inner ring curvature center of raceway groove
O_e	Outer ring curvature center of raceway groove
P	Equivalent dynamic load
P_0	Equivalent static load
Q	Rolling element load
Q_c	Rolling element load rating
q	Parameter of curvature center distance
R	Curvature center
r	Curvature radius of raceway groove
S	Probability of survival (reliability)
G_r	Radial clearance
G_a	Axial clearance
V_i	Line velocity of inner ring contact point

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V_e	Line velocity of outer ring contact point
V_c	Line velocity of rolling element center
X	Radial factor
X_0	Static radial factor
Y	Axial factor
Y_0	Static axial factor
z	Numbers of rolling elements per row
z_o	Depth of maximum orthogonal shear stress
α	Nominal contact angle
α_i	Contact angle of rolling element with inner ring
α_e	Contact angle of rolling element with outer ring
α_d	Contact angle of rolling element end with ring rib
β	Load angle
γ	Bearing structure parameter
δ	Relative approach of contact bodies
δ_e	Relative approach at outer ring raceway contact
δ_i	Relative approach at inner ring raceway contact
δ_a	Relative axial displacement between inner ring and outer ring
δ_r	Relative radial displacement between inner ring and outer ring
δ_s	Plastic deformation
η	Reduction factor of dynamic rating load
λ	Reduction factor of dynamic rating load
θ	Turn angle of axis
ν	Adjustment factor of dynamic rating load
ρ	Curvature
ρ_{I1}	curvature of body I in plane 1
ρ_{I2}	curvature of body I in plane 2
ρ_{II1}	curvature of body II in plane 1
ρ_{II2}	curvature of body II in plane 2
Σ_p	Curvature sum
$F_{(p)}$	Curvature function
ψ	Angle of rolling bearing element with maximum loaded rolling element

ψ_o	Angle of rolling bearing load range
ω	Angular speed
ω_i	Inner ring angular speed
ω_e	Outer ring angular speed
ω_w	Self rotating angular speed of rolling element

Subscripts

i	Inner ring
e	Outer ring
c	Cage
w	Rolling element
a	Axial
r	Radial
p	Central circle
m	Average
min	Minimum
max	Maximum

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Chapter 1 Introduction of Rolling Bearings

1.1 Rolling Bearing Characteristics

Rolling bearing is a key support of shafts and other rotating machine components and is one of earliest machinery components being put into large scale production. In the beginning of the 20th century, there had been emerging rolling bearing industry system in the Western countries, it had undertaken consistently the improvement of bearing design, performance and life, variety of bearing types and application, development of theoretical and calculation method. At present, the selection and calculation method for rolling bearings is perfect, product types are various, and the performance, accuracy, life and reliability could meet all requirements of all industry fields even most sophisticated field.

The friction coefficient of rolling bearings is much lower than that of sliding bearings, which is just one of some tenth or one of some hundredth of sliding bearings. It is convenient for using, maintaining, mounting and dismounting of rolling bearings. Therefore, the usage of rolling bearings is very comprehensive and in many areas rolling bearings have been progressively substituted for sliding bearings. Because of widely use and large scale production, the structures, dimensions and selection and calculation method of rolling bearings have been internationally standardized. One main task for machine design is to design the proper support arrangement and select the proper rolling bearing type according to the working condition and solve the related problems to ensure normal rotation of rolling bearings.

1.2 Rolling Bearing Classification and Suitable Scope

There are so many kinds of rolling bearings. In China National

Standards, rolling bearings are classified as 14 basic types according to the direction of load, the nominal contact angle and the rolling element shape as shown in Figure 1 – 1.

1. 2. 1 Rolling Bearing Structure Type Classification

1. Rolling bearings are classified by the direction of load and the nominal contact angle

(1) Radial rolling bearings The nominal contact angle of the radial rolling bearings which are mainly used for bearing radial load is $0^{\circ} \sim 45^{\circ}$. According to the difference of the nominal contact angle, the radial rolling bearings also can be classified as:

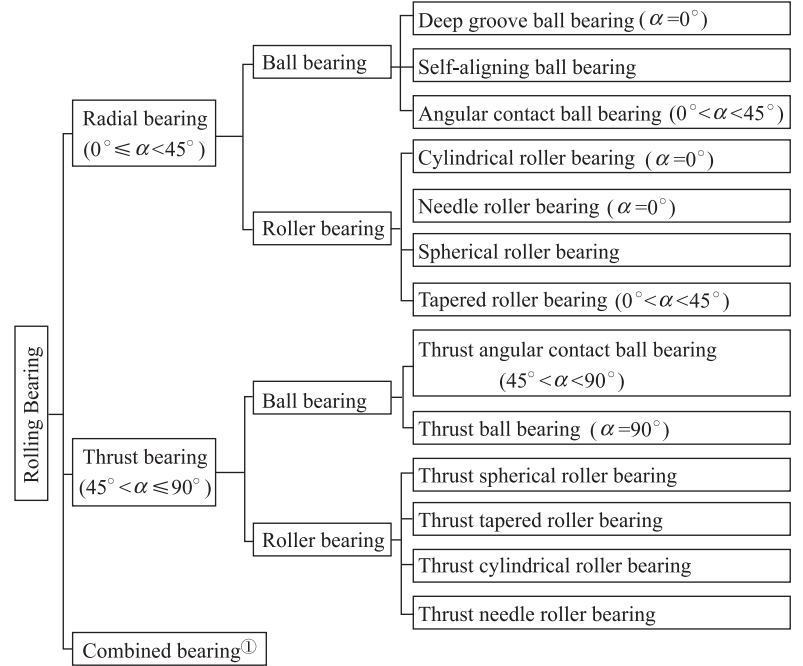


Figure 1 – 1 Rolling Bearing Classification

① A combined bearing is composed of two or more kinds of rolling bearings, such as needle roller and thrust cylindrical roller combined bearing and needle roller and thrust ball combined bearing.

1) Radial contact rolling bearings — The nominal contact angle is 0° .

2) Angular contact radial rolling bearings — The nominal contact angle is $0^\circ \sim 45^\circ$.

(2) Thrust rolling bearings The nominal contact angle is $45^\circ \sim 90^\circ$. the thrust rolling bearings also can be classified by its nominal contact angle:

1) Thrust axial contact rolling bearings — The nominal contact angle is 90° .

2) Thrust angular contact rolling bearings — The nominal contact angle is larger than 45° but smaller than 90° .

2. Rolling bearings are classified by the rolling element.

(1) Ball bearings The rolling element is ball.

(2) Roller bearings The rolling element is roller.

The roller bearings can be classified by roller type:

1) Cylindrical roller bearings — The rolling element is cylindrical roller;

2) Needle roller bearings — The rolling element is needle roller;

3) Tapered roller bearings — The rolling element is tapered roller;

4) Spherical roller bearings — The rolling element is spherical roller.

3. Rolling bearings are classified by whether it is self-aligning:

(1) Aligning rolling bearings Its raceway is spherical, which could suit the deviation of angle between the axes of two raceways and also could suit the angular motion;

(2) Non-aligning rolling bearings (rigid bearings) The non-aligning rolling bearing could prevent the deviation of angle between the axis of two raceways.

4. Rolling bearings are classified by the rows of rolling element:

(1) Single row rolling bearings The rolling bearings just have one row rolling elements;

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(2) Double row rolling bearings — The rolling bearings have two-row rolling elements;

(3) Multiple row rolling bearings — The rolling bearings have more than two row rolling elements, such as three, four row bearings.

5. Rolling bearings are classified by whether it has separable subunit.

(1) Separable rolling bearings — The subunit of the rolling bearings is separable;

(2) Non-separable rolling bearings — The subunit of the rolling bearings is not separable.

Besides, the rolling bearing could be classified by whether to have filling slot, whether to have inner ring, outer ring, ring's form, rib structure, whether to have cage and so on.

Classified by the load direction or nominal contact angle and rolling element shape, the China National Standards have stipulated 14 kinds of rolling bearings as shown in Figure 1-1.

1.2.2 Rolling Bearing Dimension Classification

Rolling bearings can be classified by its outer ring dimension as:

1) Miniature rolling bearings — The nominal outer ring dimension is under 26mm;

2) Small rolling bearings — The nominal outer ring dimension is 28 ~ 55mm;

3) Small-medium rolling bearings — The nominal outer ring dimension is 60 ~ 115mm;

4) Medium-large rolling bearings — The nominal outer ring dimension is 120 ~ 190mm;

5) Large rolling bearings — The nominal outer ring dimension is 200 ~ 430mm;

6) Special large rolling bearings — The nominal outer ring dimension is over 440mm.

1.2.3 Deep Groove Ball Bearings and Double Row Deep Groove Ball Bearings

Deep groove ball bearing is a very popular type of rolling bearings. The basic type consists of an outer ring, an inner ring, a set of balls and a set of cage, as shown in Figure 1-2. The type identification code is 6.

Deep groove ball bearings are mainly used for bearing pure radial loads and can be capable of bearing certain axial loads. When deep groove ball bearings bear pure radial loads, the contact angle is zero. If deep groove ball bearings have larger radial clearance, they would perform as angular contact ball bearings. The friction coefficient of deep groove ball bearings is very low, and the speed limit of deep groove ball bearings is very high. Therefore, deep groove ball bearings are superior to thrust ball bearings while under the circumstance of high speed and large axial load.

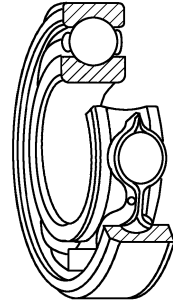


Figure 1-2 Deep groove ball bearing

The structures of deep groove ball bearings are simple. Compared with other types of rolling bearings, deep groove ball bearings are easy to manufacture in large scale and in accurate grade, and the manufacturing cost is low. So deep groove ball bearings have obtained very popular usage.

In order to meet different requirements, there are also many structural varieties, such as deep groove ball bearings with shields, deep groove ball bearings with rubber seals, deep groove ball bearings with snap ring grooves, deep groove ball bearings with filling slots for increased radial load carrying capacity, double row deep groove ball bearings.

(1) Deep groove ball bearings with shields There are two kinds of standard deep groove ball bearings with shields: Z type — one side with shield; 2Z type — two sides with shields, as shown in Figure 1-3.

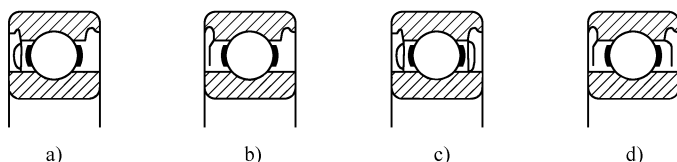


Figure 1-3 Deep groove ball bearing with shields

a) and b) Z type; c) and d) 2Z type.

The deep groove ball bearings with shields are mainly used in the applications where it is difficult to lubricate or it is not easy to set up lubrication line and inspect the lubrication condition. A certain amount of lithium based grease is filled into each of bearings by the manufacturer. Usually, the filling amount of grease is $1/4 \sim 1/3$ of the internal gap in a bearing. The filling amount of grease also can be varied by the requirement of customer. The grease could ensure bearing running under $-40 \sim 120^{\circ}\text{C}$. Customer also can demand specified grease to be filled. It is unnecessary to re-lubricate during their entire service life.

The deep groove ball bearings with shields are mostly used in medium and small generators, two sides of motor rotors, automobiles, tractors, air conditioners, electric fans and so on, and for the applications where there are requirements for low bearing noise and vibration.

(2) Deep groove ball bearings with seals There are two kinds of deep groove ball bearings with seals: bearings with contact seals RS type (one side seal) and 2RS type (two sides seals); bearing with non-contact seals RZ type (one side seal) and two RZ type (two sides seals) as shown in Figure 1-4.

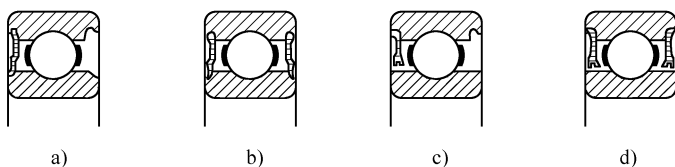


Figure 1-4 Deep groove ball bearings with seals

a) RS type b) 2RS type c) RZ type d) 2RZ type

The performance, grease filling and usage of the deep groove ball bearings with seals are basically as same as the deep groove ball bearings with shields. The differences between the bearings with seals and the bearings with shields are as follows:

The clearance between the lip of shields and the ground land of inner ring shoulder is larger for bearings with shields;

The clearance between the lip of seals and the ground land of inner ring shoulder is smaller for bearings with non-contact seals;

The clearance between the lip of seals and the ground land of inner ring shoulder is zero for bearings with contact seals. This kind of bearings have very good seal effect; but the friction is high.

For low noise applications, the small size deep groove ball bearing of 60 or 62 series should be selected. Regardless of open bearings or bearings with shields or with seals, bearings could be ordered with requirement of low noise and vibration. The China JB Standards (The China Machinery Industry Standards) have stipulated the permissible value of bearing vibration in V1, V2, V3 grades, in low, medium and high frequency bands.

(3) Deep groove ball bearings with snap ring groove and with snap ring

For the standard bearings with snap ring groove, the suffix code is N. For the bearings with snap ring groove and with snap ring, the suffix code is NR. Besides there are also many kinds of structural varieties such as ZN, ZNR and so on as shown in Figure 1-5.

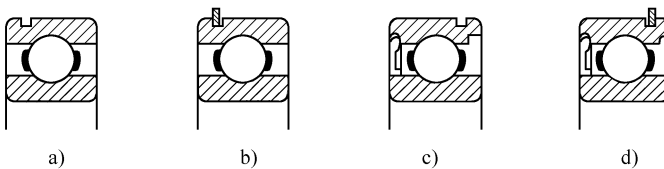


Figure 1-5 Deep groove ball bearings with snap ring groove and with snap ring

a) N type b) NR type c) ZN type d) ZNR type

Like other deep groove ball bearings, the bearings with snap ring groove and with snap ring could bear radial load, the snap rings also could confine the axial displacement of bearings, it would simplify the structure of bearing housing and shorten shaft length.

The bearings with snap ring groove and with snap ring are mostly used in automobiles and tractors where the axial load is not very large.

(4) Deep groove ball bearings with filling slots Standard deep groove ball bearings with filling slots have 200, 300 series. There is a filling slot on the same side of both inner ring and outer ring shoulders, so more balls can be inserted into one bearing. Therefore, their radial load capacity is higher than the standard deep groove ball bearings; but their axial load capacity is smaller and they can not be running at high speed.

There are many kinds of structural varieties of the bearings with filling slots as shown in Figure 1-6.

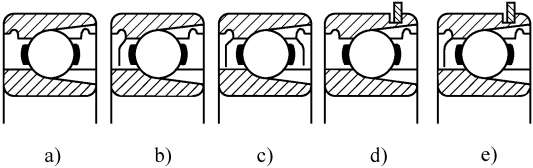


Figure 1-6 Deep groove ball bearings with filling slots

(5) Double row deep groove ball bearings The identification code of double row deep groove bearings is 4200A and 4300A. There are no filling slots for A type bearings as shown in Figure 1-7.

Two reinforced nylon cages are inserted into a bearing from both sides separately. The carrying capacity of the radial load of a double row deep groove ball bearing is 1.62 times of that of a counterpart single row bearing.

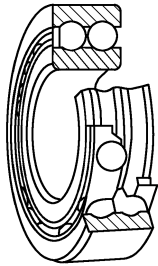


Figure 1-7 Double row deep groove ball bearings

1.2.4 Self-Aligning Ball Bearings

The most popular self-aligning ball bearings are the double row self-aligning ball bearings, it consists of one outer ring that has a large spherical groove raceway, one inner ring that has double grooves, two ball sets, and two cages. The merit of these bearings is self-aligning function that could accommodate the misalignment of two axes and shaft bending deflection. But the maximum permissible angle deviation between the axis of inner ring and the axis of outer ring should not be more than 3° as shown in Figure 1-8.

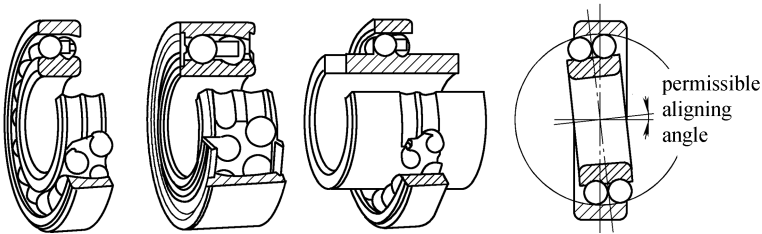


Figure 1-8 Self-aligning ball bearings

The self-aligning ball bearings are used while there is radial; load only or radial load combined with smaller axial load. Usually these bearings are not used while there only axial load, because there just one row of balls to carry axial load in that case.

There also are some types of structural varieties, such as: K type self-aligning ball bearings with tapered bore (taper ratio 1:12); Self-aligning ball bearings with tapered bore and with adapter sleeve or withdrawal sleeve; Self-aligning ball bearings with extended inner ring; Self-aligning ball bearings with thread holes for tightening screw on one side of inner ring. A self-aligning ball bearing with structural variety could be simply mounted onto the shaft directly without additional locating shoulder. The adapter sleeve may be used for adjusting the radial clearance of bearing.

Recently there has been developed the self-aligning ball bearing with double sides rubber seals, in which lithium based grease has been filled. There is no need to re-fill and maintenance free. This bearing is mainly

used in applications where there is no serious requirement for noise and vibration level and rotating speed is not high and it is not easy to mount or dismount.

There is also single row full complement self-aligning ball bearing (no cage) with seals of two sides, the bore diameter is not larger than 10mm.

Nowadays, there has been developed the reinforced structure of double row self-aligning ball bearings (the suffix code of these bearings is E). The E type bearings of bore diameter 10 ~ 65mm have comprehensively adopted the reinforced nylon 66 cages and these bearings could work effectively for a long time. In China, these bearings have already been put into production.

1.2.5 Angular Contact Ball Bearings and Double Row Angular Contact Ball Bearings

The angular contact ball bearings of 70000 type consist of outer ring, inner ring, a set of balls and cage, as shown in Figure 1-9. These bearings are particularly suitable for carrying combined radial load and axial load simultaneously and pure axial load, and they could be stably running at high speed. The single row angular contact ball bearings only could carry axial load in one direction. There an axial force component will be produced if these bearings are only carrying pure radial load, because the action line of the force of rolling elements and the action line of the radial load are not in the same radial plane but at a contact angle. So these bearings must be mounted opposed to each other in pairs.

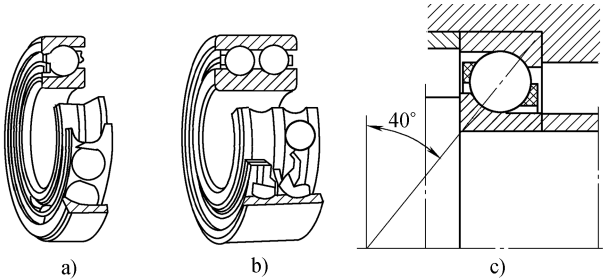


Figure 1-9 Angular contact ball bearings

a) Single row angular contact ball bearing b) Double row angular contact ball bearing c) The structure of angular contact ball bearing

1. Single row angular contact ball bearings

These bearings have the following structure types:

(1) Separable angular contact ball bearings The identification code of these bearings is S70000. The outer ring of the bearing has only one shoulder and the circular profile of the outer ring raceway merges smoothly at its apex into a straight profile. Therefore, it can be separated from its corresponding inner ring, cage and ball set. These bearings also can be mounted separately. These bearings mostly are miniature bearings whose inner bore diameter smaller than 10mm and mainly used in the applications where there require dynamic balance, low noise and vibration, high stability, such as gyroscope rotor, small electric generators, miniature motor and so on.

(2) Non-separable angular contact ball bearings This kind of bearings has a ring in which a raceway shoulder has been partly cut-off at one side, forming a locking hole between two raceway shoulders of the two rings. The two rings of the bearings can not be separated. According to the contact angles, these bearings have three types:

1) The contact angle α is equal to 40° . These bearings are suitable for carrying heavy axial loads;

2) The contact angle α is equal to 25° . these bearings are suitable for high-precision machine tool spindle operating at high speed;

3) The contact angle α is equal to 15° . Precision bearings with large dimensions, usually have a contact angle of 15° .

(3) Angular contact ball bearings in pair arrangements The angular contact ball bearings in pair arrangements are mainly suited for the applications where, combined loads of radial and axial simultaneously, and pure radial loads or axial loads in both directions are carried. These bearings are supplied to customers by bearing manufacturer according to the requirement of certain amount of preload and matching in pairs. After installation in machine and tightening, the clearance in bearings is totally eliminated, and the preload is put on the bearing pair. The stiffness of the entire bearing system is increased. There are three types of arrangements for

these bearings as shown in Figure 1 – 10.

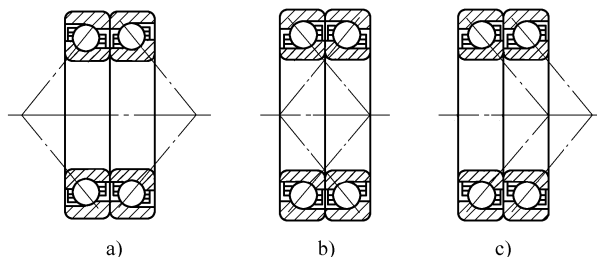


Figure 1 – 10 Pair arrangement of angular contact ball bearings

- a) Back-to-back arrangement b) Face-to-face arrangement
c) Tandem arrangement

1) Back-to-back arrangement. The suffix code is DB (such as 70000/DB). The stiffness of this arrangement is better and the performance of carrying tilting torque is good. This arrangement can carry axial loads in either directions.

2) Face-to-face arrangement. The suffix code is DF (such as 70000/DF). The stiffness and the performance of carrying tilting torque of this arrangement is not as good as the DB arrangement. But this arrangement can carry axial loads in either directions.

3) Tandem arrangement. The suffix code is DT (such as 70000DT). This arrangement could have three or more bearings in one support. But this arrangement only can carry axial load in one direction. Usually in order to balance and to limit axial displacement of a shaft, there should be third bearing to carry opposite direction axial load in another support.

Besides, there is a kind of single angular contact ball bearings that could be universally matched. These bearings are manufactured specially and could be combined as two back-to-back, two face-to-face or two tandem arrangements. The axial clearance of matching pairs could be selected according to needs. The suffix CA indicates that the axial clearance is small; the suffix CB indicates that the axial clearance is medium; the suffix CC indicates that the axial clearance is large.

They also could be paired with preload according to the requirements of applications. The suffix GA indicates that there will be small preload after matching; the suffix GB indicates that there will be medium preload after matching; the suffix GC indicates that there will be large preload after matching.

2. Double row angular contact ball bearings

The double row angular contact ball bearings are capable of carrying combined radial and axial loads, and could limit either direction displacements of shafts. The contact angle of these bearings is 32° . Comparing with the double direction thrust ball bearings, the speed limit is high and the stiffness is good. These bearings also could carry large tilting torque, widely used in front wheel hub of cars. (Some types of cars use the double row tapered roller bearings of the same size.)

There are four structural varieties for these bearings:

(1) A type This type is the standard bearing design of outer diameter smaller than or equal to 90mm. There are no filling slots in this type of bearings. They could carry either direction axial loads. In these bearings, the reinforced nylon 66 cages of light glass fiber are adopted and

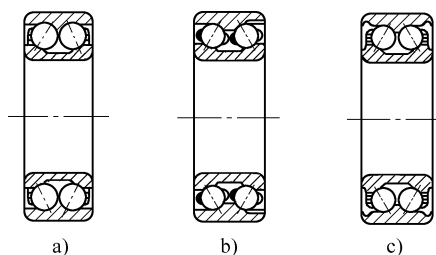


Figure 1 – 11 Double row angular contact ball bearings

a) and b) A type c) E type

the temperature rise is very low, as shown in Figure 1 – 11a.

(2) A type This type is the standard bearing design of outer diameter larger than 90mm. There is filling slot on one side of bearings. The cages are made by pressed sheet steel or brass, as shown in Figure 1 – 11b.

(3) E type This type of bearings is the reinforced design. There is filling slot on one side of bearings. More balls could be put into bearings. The load carrying capacity of this type of bearings is high, as shown in

Figure 1 – 11c.

(4) The type with shields and the type with seals on both sides of bearings. The A type design and E type design of bearings both can be with shields (non contact type) or with seals (contact type) on both sides of bearings. The bearings with shields or with seals have been filled in lithium based grease with anti-rusting agent. They can be running under temperature of $-30 \sim 110^{\circ}\text{C}$. There is no need to re-fill during their entire service life. Before installation, it should not be heated and washed. The double row angular contact ball bearings of both sides with shields and with seals are shown in Figure 1 – 12.

While installation, the side of bearing with filling slots should be opposite of the main axial load although these bearings could carry either direction axial loads.

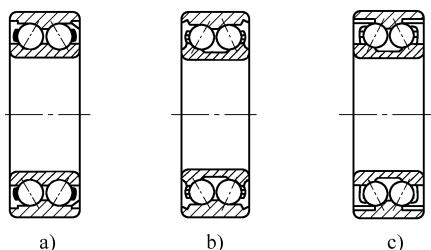


Figure 1 – 12 The double row angular contact ball bearings of both sides with shields and with seals

1. 2. 6 Single-Direction Thrust Ball Bearings and Double Direction Thrust Ball Bearings

The thrust ball bearings are separable type. The shaft washer and the housing washer can be separated from the cage and the ball set. The shaft washer is the bearing ring which is fitted with shaft. The housing washer is the bearing ring which is fitted with bearing housing and there is clearance between the housing washer and the shaft.

1. Basic type of thrust ball bearings

The basic types of thrust ball bearings are shown in Figure 1 – 13. Thrust ball bearings can be classified as: Single direction thrust ball bearings with flat housing washers having flat supporting surfaces; double direction thrust ball bearings with flat housing washers; single direction thrust ball bearings with sphered housing washers supporting surfaces; double-direction thrust ball bearings with sphered housing washers.

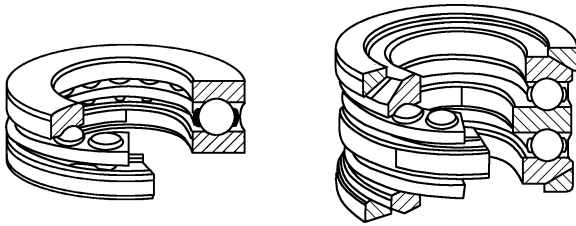


Figure 1 – 13 Thrust ball bearings

Thrust ball bearings only can carry axial loads. The single direction thrust ball bearings solely can carry one direction axial loads, and the double direction thrust ball bearings can carry either direction axial loads.

Thrust ball bearings can not prevent the radial displacement and their speed limits are very low. While the axes of the shaft and the bore of the housing washer are not in coincidence, consequently the radial runout of ball set on the raceway of the housing washer would take place and the bearing would be failed fast. In order to accomodate the error of no coincidence between axes of the shaft and the bore of the housing washer, about 0.5mm ~ 1mm clearance between the outer diameter of the housing washer and the bore of the bearing housing could be reserved. The axis of the shaft should be perpendicular with the supporting surface. In order to accomodate perpendicularity error, it is better to place elastic materials on the supporting surface of the housing washer, such as rubber, leather, varnished cloth et. If the installing condition is permissible, it may be better to select single or double direction thrust ball bearings with sphered housing washers. While thrust ball bearings rotating, there would perhaps have relative sliding between the balls and the edges of groove raceway subject to the inertia of the balls and cages. Consequently the bearing temperature would be raised. It will affect the normal rotating of the bearings. To prevent from large sliding, these bearings must always keep carrying the minimum axial load F_{amin} .

2. Double direction angular contact thrust ball bearings

The contact angle of double direction angular contact thrust ball

bearings is 60° . This type bearing consists of one housing washer of double raceway grooves with lubrication groove, two shaft washers, two cage and ball assemblies, and a central washer, as shown in Figure 1 – 14.

The height of the central washer is adjusted so that during installation the bearing is given the required clearance or preload. The amount of clearance or preload could be matched by selecting for each bearing while mounting.

Double direction angular contact thrust ball bearings are able to carry either direction axial loads, and have the advantages of high precision, excellent axial stiffness, high speed limit and low temperature rise etc. These bearings are primarily used for the spindles of modern lathes, boring machines, milling machines, and grinding machines.

1. 2. 7 Four-Point Contact Ball Bearings

Four-point contact ball bearing (QJ type) is a kind of double direction single row angular contact ball bearing, as shown in Figure 1 – 15. Due to the exceptional design of raceway grooves of inner ring and outer ring, there are two contact points between each raceway groove and a ball. The contact angle is 35° . These bearings are able to carry axial loads in double directions. The speed limits of these bearings are very high and they are specially suitable for high speed applications.

A four-point contact ball bearing is composed of one outer ring with “peach shape” section raceway groove, a pair of two-piece inner ring, and one assembly of ball set with a machined brass cage, forming four point

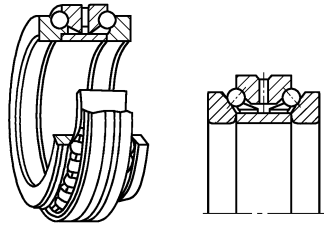


Figure 1 – 14 Double direction angular contact thrust ball bearings

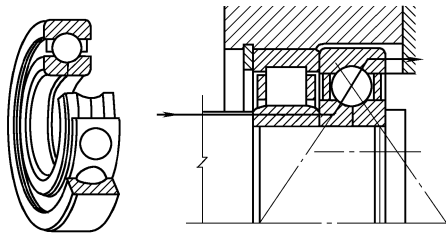


Figure 1 – 15 Four — Point Contact ball bearing

contact. The four point contact ball bearings are mainly used in the applications of machine tools, rolling mill rollers, power transmissions of large diesel engines as supports, and carrying high thrust loads. These bearings have been found extensive use in the main-shaft of gas turbine engine for ultrasonic aircrafts for carrying thrust loads acting under high accelerating condition. But there must be a precision cylindrical roller radial bearing for carrying radial loads in the same support.

It has been found by researchers that under high speed conditions, the four-point contact ball bearing must be prevented from outer ring rotating. Therefore, for these bearings which outer diameters are larger than or equal to 180mm, there have been machined two snap ring grooves with the mutuality of 180° on the mount chamfer of outer ring for preventing from outer ring rotating.

1.2.8 Insert Ball Bearings with Plummer Block Housing

Insert ball bearings with plummer block housing are one kind of bearing units, which identification code is U according to the China National Standards of rolling bearing identification code GB/T272-1993.

Insert ball bearings with housings have good aligning performance. Seal units of these bearings have double structures, as shown in Figure 1 – 16, that are compact and convenient.

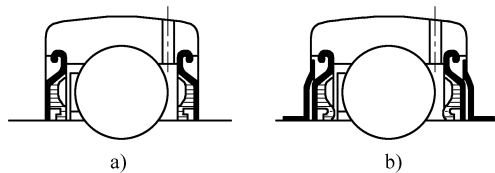
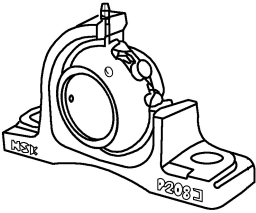
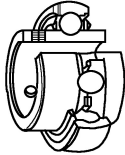
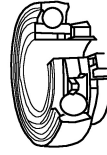
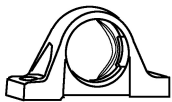
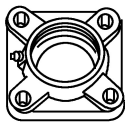


Figure 1 – 16 Seals of insert ball bearings

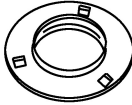
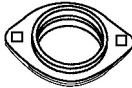
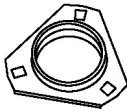
- a) Rubber seal rings on both sides of outer ring b) Metal end caps of tight fitting on both sides, having seals inside

There are many structural varieties for insert ball bearings with housings. Table 1 – 1 lists the most commonly used types of housings and insert ball bearings and their identification codes.

Table 1 – 1 The most commonly used types and their codes of housings and insert ball bearings

 Insert ball bearings with housings			The insert ball bearings inside housings		
			 With screws (UC, UB type)	 With eccentric sleeves (UE, UEL type)	 with tapered bores (UK type)
Vertical housing	Cast iron or cast steel (P)		UCP, UBP	UELP, UEP	UKP
	Pressed steel sheet (PP)		UCPP, UBPP	UELPP, UEPP	
Flanged housing	Cast iron square (FU)		UCFU, UBFU	UELFU, UEFU	UKFU

(continued)

Insert ball bearings with housings			The insert ball bearings inside housings		
			With screws (UC , UB type)	With eccentric sleeves (UE , UEL type)	with tapered bores (UK type)
Flanged housing	Cast iron diamond (FLU)		UCFLU , UBFLU	UELFLU , UEFLU	UKFLU
	Pressed steel sheet (PF)		UBPF	UELPF , UEPF	
	Pressed steel sheet diamond (PFL)		UCPFL , UBPFL	UEL PFL , UEPFL	
	Pressed steel sheet Triangle (PFT)		UCPFT , UBPFT	UFLPFT , UEPFT	
Cast iron slider housing (K)			UCK , UBK	UELK , UEK	UKK

Except that the outer diameter surface of the bearings is spherical, the others are as same as the deep groove ball bearing of the same size, and the load capacity is as same as that of the deep groove ball bearing.

1.2.9 Cylindrical Roller Bearings

Most of cylindrical roller bearings are single row cylindrical roller bearings. But there also are double row and four row cylindrical roller bearings. A single row cylindrical roller bearing consists of one outer ring, one inner ring, one cage and a set of rollers. The rollers are rotating guided by outer ring rib or inner ring rib.

For single row cylindrical roller bearings, there are the basic type, full complement type, and crossed roller type, as shown in Figure 1 – 17.

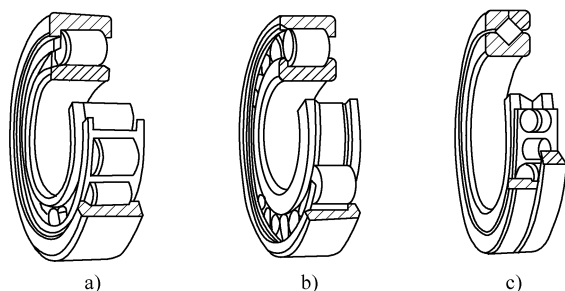


Figure 1 – 17 Cylindrical roller bearings

a) Basic type b) Full complement type c) Crossed roller type

1. Basic type of radial cylindrical roller bearings

The basic type of radial cylindrical roller bearings is separable. which consists of one outer ring or inner ring and another ring, roller, and cage assembly. According to the regulation of ISO 5753, for the P0 grade of cylindrical roller bearings, their separable rings should be interchangeable with the same type ring of the same dimensions within the international range. These bearings in general only can carry radial loads. Comparing with deep groove ball bearings of the same dimensions, their load carrying capacities are larger. But the manufacturing accuracy requirements for the matching shafts and the matching bores of housings are higher. The

permissible angle error between the axis of the shaft and the axis of the housing bore is just $2' \sim 4'$.

There are six basic types of cylindrical roller bearings for internationally interchangeable use, as shown in Figure 1 – 18:

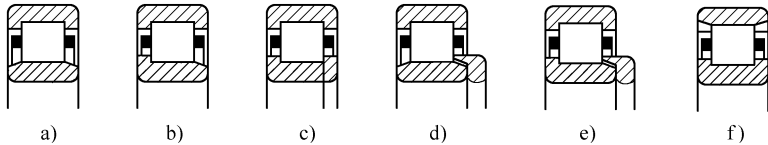


Figure 1 – 18

a) NU type b) NJ type c) NUP type d) NU + HJ type e) NJ + HJ type f) N type

1) NU type (see Figure 1 – 18a) — The inner ring with no rib can be separable from the assembly.

2) NJ type (see Figure 1 – 18b) — The inner ring with a single rib can be separable from the assembly.

3) NUP type (see Figure 1 – 18c) — The inner ring with a single rib and with a loose rib can be separable from the assembly.

4) NU + HJ type (see Figure 1 – 18d) — The inner ring with no rib but with a thrust collar can be separable from the assembly.

5) NJ + HJ type (see Figure 1 – 18e) — The inner ring with a single rib and with a thrust collar can be separable from the assembly.

6) N type (see Figure 1 – 18f) — The inner ring is with double ribs. The outer ring can be separable from the assembly.

The advantage of cylindrical roller bearings permits a few displacements of shafts. If the separable rings are without ribs, the shafts could move in double directions; if the separable rings are with a single rib, the shafts could move in the direction of no rib. Besides if both inner and outer rings are with ribs, these cylindrical roller bearings could carry certain axial loads. The axial load carrying capacities of these bearings are related to the lubrication conditions, working temperatures and heat dissipation conditions.

2. Full complement cylindrical roller bearings

Full complement cylindrical roller bearings without cage are able to be assembled with the greatest possible number of rollers so that their radial

load carrying capacities are large.

Because the contact area of full complement rollers with ring rib is increased, they can carry large axial loads. Because of no cages, there is more sliding friction between roller surfaces during bearing running, consequently lowering their speed limits.

The failure of full complement roller bearings mostly belongs to the mutual surface friction scorching of roller surfaces. Thus for the full complement roller bearings, there have been carried on the special research about chemical treatment for roller and raceway surfaces. Undergoing chemical treatment, the life of the bearings have been improved. For these bearings, their suffix code is expressed by B and B4.

3. Crossed roller bearings

The angles between the two inner ring raceways or the two outer ring raceways and the axis of the crossed roller bearing are 45° but in different directions. The two raceways alternately arrange rollers, which are crossed in right angle mutually. The roller diameter dimension is larger than the roller length, so that the cylindrical surface of a roller alone contacts the raceways of the two rings carrying the loads. Because the length of rollers is smaller than the diameter of rollers, just one end face of a roller contacts with the raceway. The end faces of rollers do not carry any loads.

The structure design of crossed roller bearings is very compact and these bearings could carry radial loads, axial loads in both directions and tilting torques. Under the radial load, the axial load and the tilting torque, the force direction and distribution condition are shown in Figure 1 – 19.

The advantages of crossed cylindrical roller bearings are as follows: large stiffness, carrying large loads, occupying little space, large bearing bore diameter, carrying very large double direction axial loads, radial loads and tilting torques.

A crossed cylindrical roller bearing consists of one inner ring, a pair of two-piece outer ring, brass cage and rollers. Cage and rollers both can be separable from rings. According to the installation location and the load distribution, the crossed cylindrical roller bearings could appropriately be

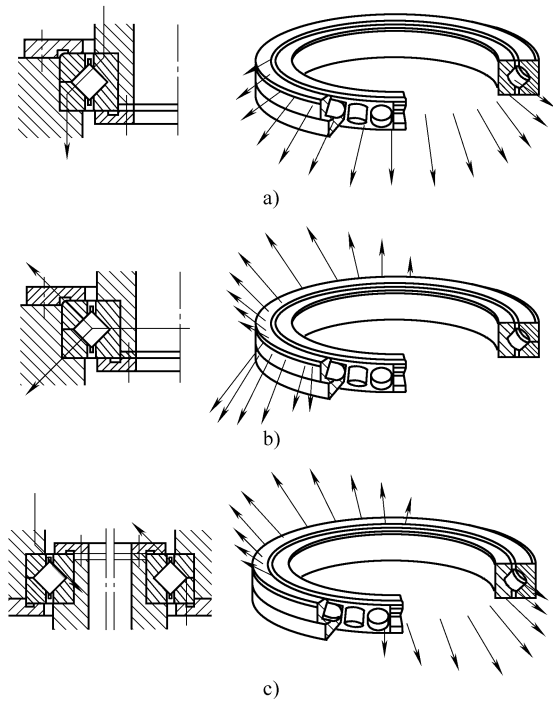


Figure 1 – 19 The crossed cylindrical roller bearings

- a) the transmission of axial load b) the transmission of radial load
c) the transmission of tilting torque

designed as many structure types such as single row, double row, full complement rollers or cage et cetera.

1. 2. 10 Tapered Roller Bearings

Most of the tapered roller bearings are single row tapered roller bearings. Recently the double-row tapered roller bearings of small size are used in front hubs of small cars. The four-row tapered roller bearings are predominantly used in large scale cold and hot rolling mills. Figure 1 – 20 shows the standard structures of single row, double row and four row tapered roller bearings.

A single row tapered roller bearing comprises an outer ring, an inner ring and a tapered roller set retaining as an assembly by a “basket” shape

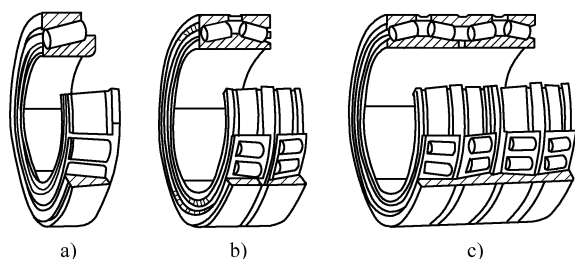


Figure 1-20 Tapered roller bearings

a) single row b) double row c) four row

cage. The outer ring can be separated from the inner ring assembly. According to the ISO stipulation of the outside dimensions of tapered roller bearings, any standard outer ring or inner ring assembly of standard type tapered roller bearings should be inter changeable with the same type outer ring or inner ring assembly within the international range. That is, not only the outside dimensions and the tolerance of the same type bearings should conform with the ISO 492 (GB 307) stipulation, the taper angle of inner ring assembly and the assembly cone diameter should conform with the related inter changeable stipulations as well.

Usually, the taper angles of outer raceways of single row tapered roller bearings are within $10^{\circ} \sim 19^{\circ}$, which can carry the combined loads of radial and axial loads at the same time. The larger the taper angle is, the larger the axial load capacity is. For the bearings with larger taper angles, their suffix code is B and their taper angles are within $25^{\circ} \sim 29^{\circ}$, which can carry larger axial loads. Moreover, the clearances of single row tapered roller bearings can be adjusted during installation so that an optimum clearance or preload for each application can be achieved.

The outer rings (or inner rings) of double row tapered roller bearings are an integral. The small end faces of the two inner rings (or outer rings) are near each other, in between which there is a spacer. Its height could be used for adjusting clearance or preload.

1.2.11 Spherical Roller Bearings and Thrust Spherical Roller Bearings

1. Spherical roller bearings (21200, 22200, 23200, ... types)

A spherical roller bearing consists of one double raceway inner ring, two rows of spherical rollers, a cage and one outer ring with a large spherical raceway. Spherical roller bearings possess self-aligning performance as shown in Figure 1-21. Spherical roller bearings can carry double direction axial loads, and also can carry large radial loads at the same time. There are two types of inner ring bores of the spherical roller bearings: cylindrical bore and tapered bore. If the inner ring bores of the spherical roller bearings are tapered bores, these bearings can be mounted on the adapter sleeves or the withdrawal sleeves, as shown in Figure 1-22, and these bearings could be mounted on any position of a shaft with cylindrical surface of uniform diameter or on a stepped shaft. The spherical roller bearings with adapter sleeves also could be used for adjusting radial bearing clearance.

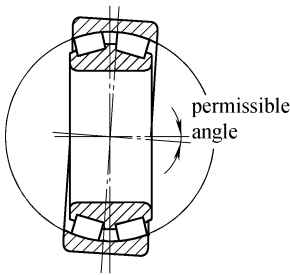


Figure 1-21 Spherical roller bearings

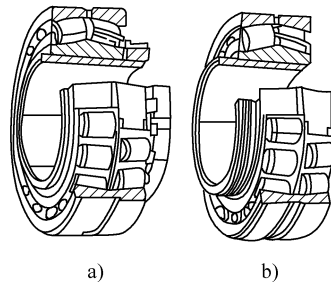


Figure 1-22 Spherical roller bearings with adapter sleeves and withdrawal sleeves
a) with adapter sleeves b) with withdrawal sleeves

(1) The structure types of spherical roller bearings

1) C, CC and EC types are shown in Figure 1-23a. These bearings have two rows of the convex, symmetrical, spherical rollers. There are no ribs in the inner ring. But there is a guide ring and have two pressed steel sheet cages. The guide ring rotates under the inner ring guiding. The EC type of the bearings has the enforced convex, symmetrical, spherical rollers, so that these bearings can carry larger loads. The rollers and

raceways of the CC type bearings have been through optimizing processes, which are helpful for roller guiding and reducing friction heat.

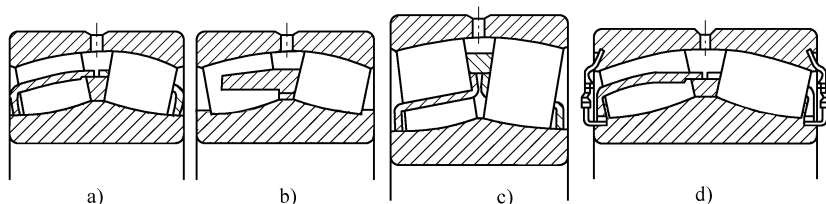


Figure 1 – 23 The structure types of spherical roller bearings

- a) C, CC, EC types b) CA, CAC and ECA types c) E type
d) Spherical roller bearings with seals

2) CA、CAC、ECA types are shown in Figure 1 – 23b. These types of bearings have two rows of the convex, symmetrical, spherical rollers. There are two small ribs in the inner ring and there is a guide ring rotated by the inner ring guiding, and the guide ring also guides the two rows of convex, symmetrical, spherical rollers rotating. The dimensions of these bearings are large, and these bearings have small ribs as well as a machined brass cage rotated by the guide ring. The rollers and raceways of CA, CAC type bearings have been through optimizing processes, which are helpful for roller guiding and reducing friction heat.

3) E type is shown in Figure 1 – 23c. This type of bearings is an enforceable type, which has an increase of the load carrying capacity by increasing the roller diameter.

The sealing type of spherical roller bearings is shown in Figure 1 – 23d.

4) The spherical roller bearings of vibration screens. This kind of spherical roller bearings is shown in Figure 1 – 24. Working under large vibration, the radial clearance of these bearings is required to be large, generally between C3 and C4. The bearings adopt the pressed steel cage, which is hardened by surface heat treatment that improves wear resistance. The inner ring of the bearings with large dimensions has no ribs. The guide ring is located on the outer raceway, which is rotated by the guiding of the outer raceway. At the same time, a layer of PTFE is lined on the inner

surface of the cylindrical bore, which can reduce vibration and has corrosion resistance.

(2) Angle error of spherical roller bearings

Spherical roller bearings have self-aligning performance. They can automatically adjust the angle error caused by not good installation. The adaptability to angle error of the spherical roller bearings changes with width series and diameter series. If the inner diameter is the same, then the wider the width is, the larger the outside diameter is, and the larger the permissible angle error is.

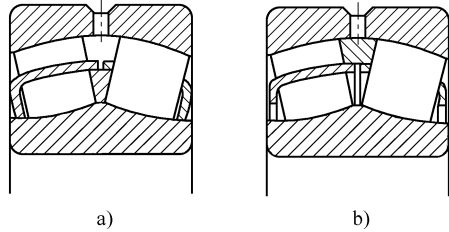


Figure 1-24 Spherical roller bearings of vibration screens

- a) bearing inner ring diameter $d < 75\text{mm}$
- b) bearing inner ring diameter $d \geq 75\text{mm}$

- 1) For the 213 series, the permissible angle error is 1° .
- 2) For the 222 series, the permissible angle error is 1.5° .
- 3) For the 223 series, the permissible angle error is 2° .
- 4) For the 230 series, the permissible angle error is 1.5° .
- 5) For the 231 series, the permissible angle error is 1.5° .
- 6) For the 232 series, the permissible angle error is 2.5° .
- 7) For the 239 series, the permissible angle error is 1.5° .
- 8) For the 240 series, the permissible angle error is 2° .
- 9) For the 241 series, the permissible angle error is 2.5° .

2. Thrust spherical roller bearings

A thrust spherical roller bearing consists of a shaft washer, a housing washer and a set of asymmetrical rollers. In thrust spherical roller bearings, the load is transmitted from one raceway to another raceway at an angle (the contact angle between the load action line of rollers and the axial line of bearing). They can therefore support larger axial load and simultaneously support radial load and have the important capability of self-aligning, as shown in Figure 1-25.

Thrust spherical roller bearings are separable type and have a pressed steel cage. that with the spherical rollers, forms a non-separable assembly with a shaft washer. A housing washer can be separable with this assembly. They are suitable for using in relatively high speed working conditions, mostly used in spindles of petroleum drilling rigs.

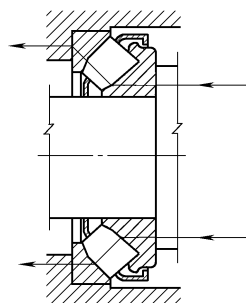


Figure 1 – 25 Thrust spherical roller bearing

1. 2. 12 Needle Roller Bearings

A needle roller bearing is composed of a ring, needle rollers and a cage. By comparison with other types of bearings, for the same load carrying capacity, the cross section dimension of needle roller bearings is the smallest. Needle roller bearings with a cage are especially suitable for use in the applications of high speeds, heavy loads and small permissible space. The needle roller can be called needle form roller, which form is slender and which outside diameter usually is not larger than 4mm. The end shape of the needle rollers can be circular, sharp taper or concave arc. The commonly used needle roller bearings have the following structure types as shown in Figure 1 – 26:

- 1) Drawn cup needle roller bearings (see Figure 1 – 26 a and b).
- 2) Machined ring needle roller bearings (see Figure 1 – 26 c and d).

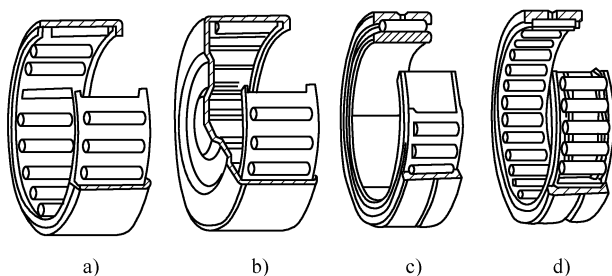


Figure 1 – 26 needle roller bearings

- a) and b) drawn cup needle roller bearing
c) and d) machined ring needle roller bearing

- 3) Combined needle roller bearings.
- 4) Needle rollers and cage assembly.
- 1. Drawn cup needle roller bearings

These drawn-cup needle roller bearings have two types: across type (HK type) and close type (BK type). The rings of the across type needle roller bearings have a thin-wall ring and are made by the material of 1mm thickness of low carbon steel or low carbon alloy steel after through many times of drawn-cup, bending and chemical heat treatment. Therefore, the cross-section dimension of needle roller bearings is the smallest. There are no inner rings for the drawn-cup needle bearings. The needle rollers are in direct contact with the shaft surface so that the shaft surface contacted with the needle rollers should be manufactured with the same accuracy and hardness of heat treatment as that of the bearing inner rings. The drawn-cup needle roller bearings with cages are suitable for use in applications of high speeds and heavy loads, such as transmissions of machine tools, automobiles, motorcycles or textile machines. The drawn-cup needle roller bearings without cages are full complement bearings. These bearings are suitable for use in swing conditions under heavy loads, such as shaft ends of airframes and main wings of missiles.

2. Needle roller bearings with machined rings

The outside dimensions and the structures of the needle roller bearings with machined rings are determined according to China National Standard GB/T 5801. Commonly used needle roller bearings with machined rings are NA type or RNA type. The NA type needle roller bearings are composed of a machined outer ring, an inner ring, a nylon cage or pressed steel cage and a set of rollers with a round end. The machined outer ring and inner ring are made by bearing steel through hardening and grinding. The outer rings have double ribs and the inner rings have no ribs (see the lower part of Figure 1-27). The inner ring can be separable from the assembly of the outer ring with a set of needle rollers. The RNA type needle roller bearings are needle roller bearings without inner ring (see the upper part of Figure 1-27). Because the shaft journal surface contacts directly with the needle rollers, the shaft journal surface should be manufactured with the same accuracy and

hardness of heat treatment as that of the bearing inner rings.

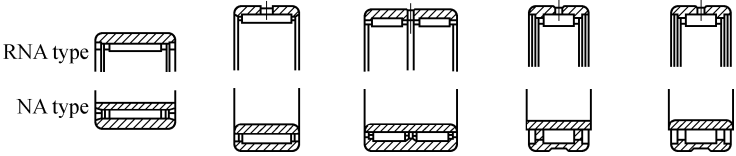


Figure 1-27 Needle roller bearing with machined rings

The needle roller bearings with machined rings mostly are used in transmissions of machine tools and automobiles. The advantages of these bearings are small cross-section dimension and high radial load capacity.

The non-standard full complement needle roller bearings with machined rings have found their wide applications in swing motion and carrying large load conditions, such as locomotive rods and airplane tail parts. Figure 1-28 shows a three-ring needle roller bearing that could automatically adjust the angle deviation by the joint bearing part.

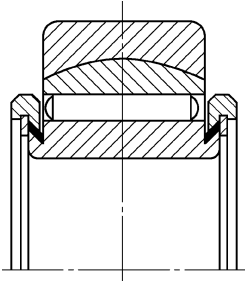


Figure 1-28 The e-ring needle roller bearing

3. Combined needle roller bearings

A combined needle roller bearing is a combination of needle roller bearing with machined rings, ball bearings and roller bearings. Usually the bearings have the following structure types: (see Figure 1-29)

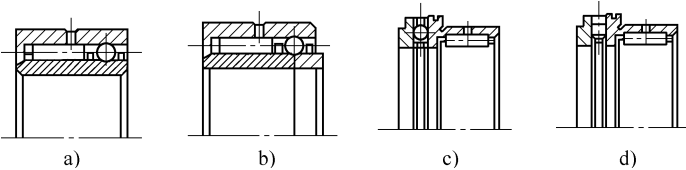


Figure 1-29 Combined needle roller bearings

- (1) NKIA type — Needle roller bearing and angular contact radial ball bearing are combined.
- (2) NKIB type — Needle roller bearing and separable angular contact

radial ball bearing are combined.

(3) NKX type — Needle roller bearing without inner ring and thrust ball bearing are combined.

(4) NKXR type — Needle roller bearing without inner ring and thrust roller bearing are combined.

The NKIA type combined bearings are suitable for the applications where the installation space is limited, the rotation speed is high and carrying double direction axial loads except carrying radial load is required. These bearings have an axial clearance about 0.1 ~ 0.3mm.

The NKIB type combined bearing is a four-point contact ball bearing except the above mentioned characteristics.

The NKX type combined bearings are needle roller bearings without inner rings, the technical requirements for shaft are high. In addition to carrying radial load, these bearings can carry single direction axial load.

The NKXR type combined bearings have the similar characteristics as the NKX type combined bearings. But they can carry even much larger single direction axial load.

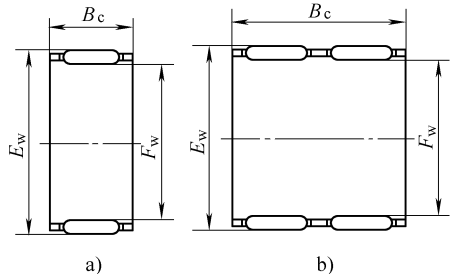


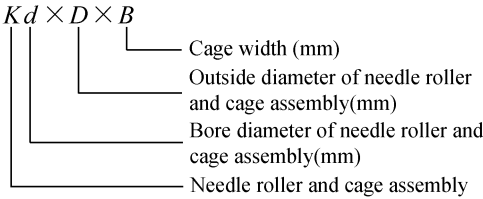
Figure 1-30 Needle roller and cage assembly

a) K type b) K...ZW type

4. Needle roller and cage assembly

Needle roller and cage assembly is actually a needle roller bearing without rings, as shown in Figure 1-30.

The international expression of the identification code of needle roller and cage assembly is as follows:



The needle roller and cage assembly is the thinnest rolling bearing. The surface of the bore of housing functions as its outside raceway and the shaft surface functions as its inner raceway during rotating or swing movement. The needle roller and cage assembly can carry impulse loads and alternating loads and can be used in gear boxes and large and small end of connecting links in motorcycles and automobiles. Because the temperature is high in the cylinder, the bearings should endure high temperature.

The needle roller and cage assembly can use the pressed steel cage or machined brass cage. But the cost of these two cages are high. At present, the smallest cages are molding cages of reinforced glassfiber nylon 66. The cost of these cages is low and the quality is high.

1. 2. 13 Thrust Roller Bearings

Thrust roller bearings are mainly used for carrying heavier axial loads, taking small space and having higher stiffness, mostly used in heavier engineering machines, such as petroleum drilling rigs. A thrust roller bearing consists of a shaft washer, a housing washer and one assembly of roller set and cage. The three parts are separable. The single direction thrust roller bearing can only carry single direction axial load, as shown in Figure 1-31.

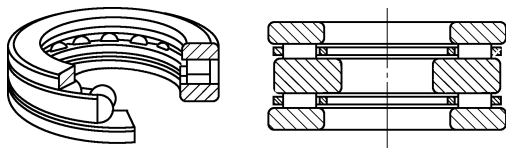


Figure 1-31 Thrust roller bearing

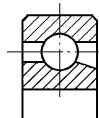
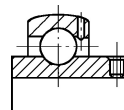
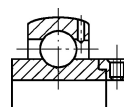
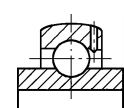
In thrust roller bearings, most of the cages are molding cages using reinforced glassfiber nylon 66. But these cages are suitable for working conditions under 120°C.

A thrust roller bearing could be without shaft washer and housing washer, only having an assembly of roller set and cage. But in this situation, the surfaces of the related parts contacting with rollers must be machined as same as the bearing raceway in order to guarantee the normal working of a machine.

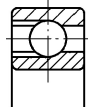
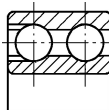
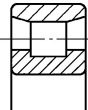
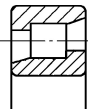
1. 2. 14 Commonly Used Rolling Bearings Structure Type Classification

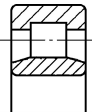
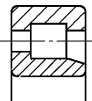
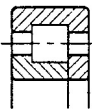
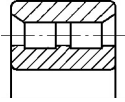
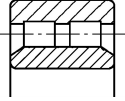
The commonly used rolling bearing structure type classification is shown in Table 1-2.

Table 1-2 Commonly used rolling bearing structure types

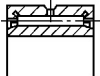
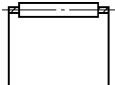
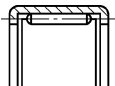
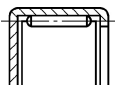
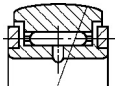
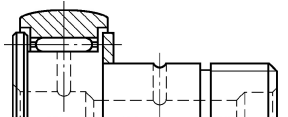
bearing type classification								name		sketch		type code ^①		standard numbers		
radial bearing	radial contact bearing	radial contact ball bearing	deep groove ball bearing	single row	non-separable	Without	Without filling slot					6		GB/T 276		
								spherical outside surface	with threaded pin	insert (deep groove) ball bearing with threaded pin			UC		GB/T 3882	
									with eccentric self-locking collar	insert (deep groove) ball bearing with eccentric self-locking collar			UEL			
									tapered bore	tapered bore insert (deep groove) ball bearing			UK			

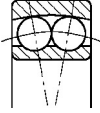
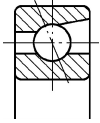
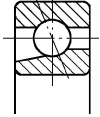
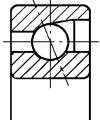
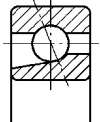
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bearing type classification							name	sketch	type code ^①	standard numbers		
radial bearing	radial contact bearing	radial contact ball bearing	deep groove ball bearing	single row	non-separable	with filling slot		deep groove ball bearing with filling slot and cage		6 ^②	—	
				double row				double row deep groove ball bearing		4	—	
			cylindrical roller bearing	single row		outer ring without rib	inner ring with double ribs	cylindrical roller bearing		N	GB/T 283	
						outer ring with single rib		cylindrical roller bearing with outer ring with single rib		NF		
			cylindrical roller bearing	single row								

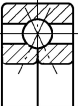
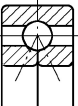
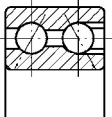
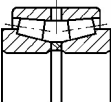
bearing type classification										name	sketch	type code ^①	standard numbers
radial bearing	radial contact bearing	radial contact ball bearing	cylindrical roller bearing	single row	separable	outer ring with double ribs	inner ring without rib	without rib	cylindrical roller bearing with inner ring without rib		NU	GB/T 283	
							inner ring with single rib		cylindrical roller bearing with inner ring with single rib		NJ		
							with loose rib	cylindrical roller bearing with inner ring with single rib and loose rib		UNP			
				double rows		outer ring without rib	inner ring with double ribs	double row cylindrical roller bearing with outer ring without rib		NN	GB/T 285		
						outer ring with double ribs	inner ring without rib	double row cylindrical roller bearing with inner ring without rib		NNU			

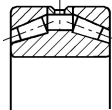
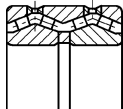
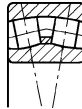
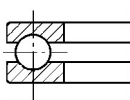
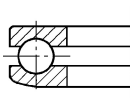
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bearing type classification							name		sketch		type code ^①		standard numbers	
radial bearing	radial contact bearing	radial contact roller bearing	needle roller bearing	single row	—	outer ring with double ribs	inner ring without rib	needle roller bearing		NA	GB/T 5801			
						without outer ring		without inner ring	needle roller and cage assembly (single row radial contact)		K	JB/T 7918		
						drawn cup	through bore		through bore type drawn cup (single row radial) needle roller bearing		HK	GB/T 290		
							close end type	close end type drawn cup (single row radial) needle roller bearing		BK				
						track roller outer ring without rib	inner ring with loose rib	track needle roller bearing (single row) with loose rib			NATR	GB 6645		
							stud type track roller	stud type track needle roller bearing (single row)			KR			

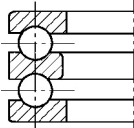
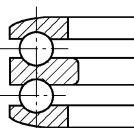
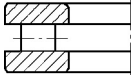
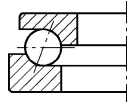
								(continued)	
bearing type classification						name	sketch	type code ^①	standard numbers
radial bearing	angular contact radial bearing	angular contact radial ball bearing	self-aligning ball bearing	double rows	outer ring with spherical raceway	(double row radial) aligning ball bearing		1	GB/T 281
			angular contact ball bearing	non-separable	counter bored outer ring	(counter bored outer ring single row radial) angular contact ball bearing		7	GB/T 292
					stepped inner ring	stepped inner ring (single row radial) angular contact ball bearing		B7	
					separable outer ring	separable outer ring (single row radial) angular contact ball bearing		S7	
					separable inner ring	separable inner ring (single row radial) angular contact ball bearing		SN7	
				single row	separable				

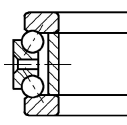
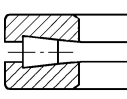
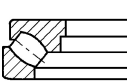
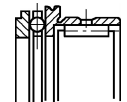
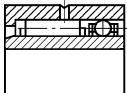
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bearing type classification								name	sketch	type code ^①	standard numbers
radial bearing	angular contact radial bearing	angular contact radial ball bearing	angular contact ball bearing	single row	separable	two-piece inner ring	four-point-contact	(two-piece inner ring single row radial) four-point-contact ball bearing		QJ	GB/T 294
							three-point-contact	(two-piece inner ring single row radial) three-point contact ball bearing		QJS	
		angular contact radial roller bearing	tapered roller bearing	double rows	non-separable	with filling slot		double row (radial) angular contact ball bearing		0 ^②	GB/T 296
				single row	separable	—		(single row radial) tapered roller bearing		3	GB/T 297
				double rows		double-inner ring		double row (radial) tapered roller bearing		35	GB/T 299

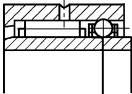
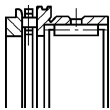
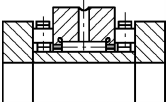
bearing type classification										name	sketch	type code ^①	standard numbers
radial bearing	angular contact radial bearing	angular contact roller bearing	tapered roller bearing	double rows	separable	double outer ring		double row (radial) tapered roller bearing		37	GB/T 299		
			four rows	double-inner ring		four row (radial) tapered roller bearing		38	GB/T 300				
			self-aligning roller bearing	double rows	non-separable	outer ring with spherical raceway		double row (radial) aligning roller bearing		2	GB/T 288		
thrust bearing	axial contact bearing	axial contact ball bearing	thrust ball bearing	single row	separable	single direction	plane type	(single direction) thrust ball bearing		5	GB/T 301		
							spherical surface type	spherical surface type (single direction) thrust ball bearing					

(continued)

bearing type classification							name	sketch	type code ^①	standard numbers
thrust bearing	axial contact bearing	axial contact ball bearing	thrust ball bearing	double rows	separable	double direction	plane type double row thrust ball bearing		5	GB/T 301
							spherical surface type spherical surface type double row thrust ball bearing			
		axial contact ball bearing	thrust cylindrical roller bearing	single row	—	single direction	plane type (single direction) thrust cylindrical roller bearing		8	GB/T 4663
			thrust needle roller bearing				without washer (single direction) thrust needle roller and cage assembly			
	angular contact thrust bearing	angular contact thrust ball bearing	thrust angular contact ball bearing		separable		plane type (single direction) thrust angular contact ball bearing		56	

bearing type classification										name	sketch	type code ^①	standard numbers
thrust bearing	angular contact bearing	angular contact thrust ball bearing	thrust angular contact ball bearing	double row	separable	double direction	plane type	double direction thrust angular ball bearing		23	JB/T 6362		
		angular contact thrust roller bearing	thrust tapered roller bearing	single row				single direction	plane type	(single direction) thrust tapered roller bearing		9	—
			thrust aligning roller bearing							(single direction) thrust aligning roller bearing		2	GB/T 5859
	combined bearing					single direction	radial needle roller	thrust ball	(single row radial) needle roller and thrust ball combined bearing		NKX	JB 3122	
								angular contact thrust ball	(single row radial) needle roller and angular contact ball combined bearing		NKIA	JB 2123	

(continued)

bearing type classification					name	sketch	type code ^①	standard numbers
combined bearing	separable type	double directions	radial needle roller	angular contact thrust ball	(single row radial) needle roller and double direction angular contact ball combined bearing		NKIB	JB 2123
		single direction		thrust cylindrical roller	(single row radial) needle roller and (single direction) thrust cylindrical roller combined bearing		NKXR	ZB J11011
		double direction			(single row radial) needle roller and double direction thrust cylindrical roller combined bearing		ZARN	JB/T 6644

① Usually the type code is omitted in the bearing identification code and is not expressed.

1.3 Identification Code of Rolling Bearing

1.3.1 New Rolling Bearing Identification Code in China

The Chinese new rolling bearing identification code standards (GB/T272—1993 , JB/T 2974—2004) have been put into practice since the first of July , 1994 , In these new standards , each rolling bearing is designated by an identification code that consists of an English letter and a number that clearly indicates bearing construction , dimension , tolerance grade and technical characteristics.

1. Rolling bearing identification code

A rolling bearing identification code consists of three parts as basic code , prefix code and suffix code . The arrangement order of each part is shown in Table 1 - 3 .

Table 1 - 3 Structure of rolling bearing identification code

bearing identification code													
prefix code	basic code					suffix code							
bearing sub parts	structure type	dimension series		bore diameter	contact angle	1	2	3	4	5	6	7	8
		width (height) series	diameter series			inner structure	seal or shield	cage and its material	bearing material	tolerance grade	clearance	arrangement	others

2. Basic code of rolling bearing (except needle roller bearing)

(1) Bearing type code Bearing type code consists of numbers and letters as shown in Table 1 - 4 .

Table1 - 4 Bearing type code

code	bearing type	code	bearing type
0	double row angular contact bearing	N	cylindrical roller bearing
1	self-aligning ball bearing		double row or more row expressed by letter NN
2	self-aligning roller bearing and thrust self-aligning roller bearing	U	insert ball bearing

(continued)

code	bearing type	code	bearing type
3	tapered roller bearing	QJ	Four-point contact ball bearing
4	double row deep groove ball bearing		
5	thrust ball bearing		
6	deep groove ball bearing		
7	angular contact ball bearing		
8	thrust cylindrical roller bearing		

(2) Dimension series code

Bearing dimension series code consists of bearing width (height) series code and diameter series code. The dimension series codes of radial bearing and thrust bearing are shown in Table 1 - 5.

Table 1 - 5 Bearing dimension series code

Diameter series code	radial bearing								thrust bearing			
	width series code								height series code			
	8	0	1	2	3	4	5	6	7	9	1	2
	dimension series code											
7	—	—	17	—	37	—	—	—	—	—	—	—
8	—	08	18	28	38	48	58	68	—	—	—	—
9	—	09	19	29	39	49	59	69	—	—	—	—
0	—	00	10	20	30	40	50	60	70	90	10	—
1	—	01	11	21	31	41	51	61	71	91	11	—
2	82	02	12	22	32	42	52	62	72	92	12	22
3	83	03	13	23	33	—	—	—	73	93	13	23
4	—	04	—	24	—	—	—	—	74	94	14	24
5	—	—	—	—	—	—	—	—	—	95	—	—

(3) Bearing bore diameter code

The bearing bore diameter codes are shown in Table 1 - 6.

Table 1 – 6 Bearing bore diameter code

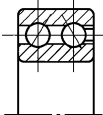
Nominal bearing bore diameter/mm		bore diameter code	example
0. 6 to 10 (not integer)		expressed directly by the nominal bore diamer number of mm, between bore diameter code and dimension series code divided by “/”.	deep groove ball bearing 618/2.5 $d = 2.5\text{mm}$
1 to 9 (integer)		expressed directly by the nominal bore diameter number of mm, for deep groove ball bearing and angular contact ball bearing 7, 8 and 9 diameter series; between bore diameter code and dimension code divided by “/”.	deep groove ball bearing 62 5 618/5 $d = 5\text{mm}$
10 to 17	10 12 15 17	00 01 02 03	deep groove ball bearing 62 00 $d = 10\text{mm}$
20 to 480 (except 22, 28, 32)		nominal bore diameter by divding by five; if the quotient is a single digit number, before the quotient should add “0”.	aligning roller bearing 232 08 $d = 40\text{mm}$
≥ 500 and 22, 28, 32		directly expressed by bore diameter number of mm; but between bore diameter and dimension series code divided by “/”.	Aligning roller bearing 230/500 $d = 500\text{mm}$ deep groove ball bearing 62/22 $d = 22\text{mm}$

For example: aligning roller bearing 23224:2 — type code, 32 — dimension series code, 24 — bore diameter code, $d = 120\text{mm}$.

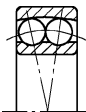
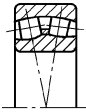
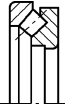
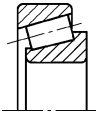
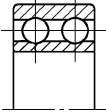
(4) Commonly used bearing series code

Bearing combination code consists of commonly used bearing structure type code and dimension series code, as shown in Table 1 – 7.

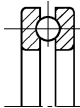
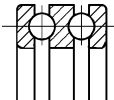
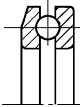
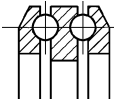
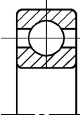
Table 1 – 7 Commonly used bearing combination code

bearing type	sketch	type code	dimension series code	bearing combination code	standard number
double row angular contact ball bearing		(0)	32	32	GB/T 296
		(0)	33	33	

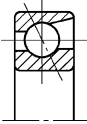
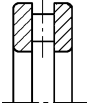
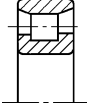
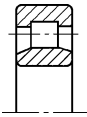
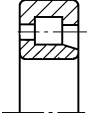
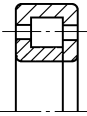
(continued)

bearing type	sketch	type code	dimension series code	bearing combination code	standard number
aligning ball bearing		1 (1) 1 (1)	(0) 2 22 (0) 3 23	12 22 13 23	GB/T 281
aligning roller bearing		2 2 2 2 2 2 2	13 22 23 30 31 32 40 41	213 222 223 230 231 232 240 241	GB/T 288
thrust aligning roller bearing		2 2 2	92 93 94	292 293 294	GB/T 5859
tapered roller bearing		3 3 3 3 3 3 3 3 3 3	02 03 13 20 22 23 29 30 31 32	302 303 313 320 322 323 329 330 331 332	GB/T 297
double row deep groove ball bearing		4 4	(2) 2 (2) 3	42 43	

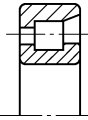
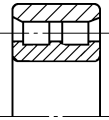
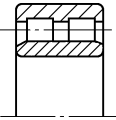
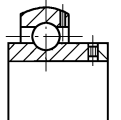
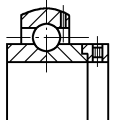
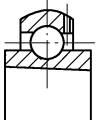
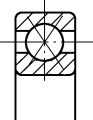
(continued)

bearing type		sketch	type code	dimension series code	bearing combination code	standard number
thrust ball bearing	thrust ball bearing		5	11	511	GB/T 301
			5	12	512	
			5	13	513	
			5	14	514	
	double direction thrust ball bearing		5	22	522	GB/T 301
			5	23	523	
			5	24	524	
	thrust ball bearing with spherical surface housing		5	32 ^①	532	
			5	33	533	
			5	34	534	
	double direction thrust ball bearing with spherical sur- face housings		5	42 ^②	542	
			5	43	543	
			5	44	544	
deep groove ball bearing			6	17	617	GB/T 276 GB/T 4221
			6	37	637	
			6	18	618	
			6	19	619	
			16	(0) 0	160	
			6	(1) 0	60	
			6	(0) 2	62	
			6	(0) 3	63	
			6	(0) 4	64	

(continued)

bearing type		sketch	type code	dimension series code	bearing combination code	standard number
angular contact ball bearing			7	19	719	GB/T 292
			7	(1) 0	70	
			7	(0) 2	72	
			7	(0) 3	73	
			7	(0) 4	74	
thrust cylindrical roller bearing			8	11	811	GB/T 4663
			8	12	812	
cylindrical roller bearing	cylindrical roller bearing with outer ring without rib		N	10	N10	GB/T 283
			N	(0) 2	N2	
			N	22	N22	
			N	(0) 3	N3	
			N	23	N23	
			N	(0) 4	N4	
	cylindrical roller bearing with inner ring without rib		NU	10	NU 10	
			NU	(0) 2	NU 2	
			NU	22	NU 22	
			NU	(0) 3	NU 3	
			NU	23	NU 23	
			NU	(0) 4	NU4	
	cylindrical roller bearing with inner ring with single rib		NJ	(0) 2	NJ2	
			NJ	22	NJ22	
			NJ	(0) 3	NJ3	
			NJ	23	NJ23	
			NJ	(0) 4	NJ4	
	cylindrical roller bearing with inner ring with single rib and loose rib		NUP	(0) 2	NUP 2	
			NUP	22	NUP 22	
			NUP	(0) 3	NUP 3	
			NUP	23	NUP 23	

(continued)

bearing type		sketch	type code	dimension series code	bearing combination code	standard number
cylindrical roller bearing	cylindrical roller bearing with outer ring with single rib		NF	(0) 2 (0) 3 23	NF2 NF3 NF23	GB/T 283
	double row cylindrical roller bearing		NN	30	NN 30	GB/T 285
	double row cylindrical roller bearing with inner ring with out rib		NNU	49	NNU 49	
insert ball bearing	insert ball bearing with threaded pins		UC UC	2 3	UC 2 UC 3	GB/T 3882
	insert ball bearing with eccentric self-locking collars		UEL UEL	2 3	UEL 2 UEL 3	
	insert ball bearing with tapered bore		UK UK	2 3	UK 2 UK 3	
four-point-contact ball bearing			QJ	(0) 2 (0) 3	QJ 2 QJ 3	GB/T 294

Note: In the table, the numbers by “ () ” can be omitted in the combination code.

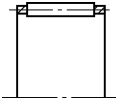
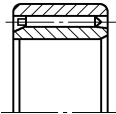
① Dimension series actual as 12, 13 and 14, are expressed by 32, 33 and 34, respectively.

② Dimension series actual as 22, 23 and 24, are expressed by 42, 43 and 44, respectively.

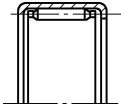
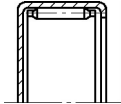
3. The needle roller bearing basic code

Basic code of needle roller bearing consists of the bearing type code and the feature dimension code expressing bearing fit and installation. The expressing methods of needle roller bearing code are shown in Table 1 – 8.

Table 1 – 8 Needle roller bearing basic code

Bearing type		sketch	type code	feature dimension code of expressing bearing fits and installation		Bearing basic code	standard number
needle roller and cage assembly	needle roller and cage assembly		K	$F_w * E_w$ $* B_c$		$KF_w * E_w$ $* B_c$	GB/T 5846
	thrust needle roller and cage assembly		AXK	$D_{cl} D_c^{①}$		$AXKD_{cl} D_c$	GB/T 4605
needle roller bearing	needle roller bearing		NA	expressing by dimension series code		$NA\ 4800$ $NA\ 4900$ $NA\ 6900$	GB/T 5801
				dimension series code 48 49 69	② bore diameter code in table 1 – 6		

(continued)

Bearing type		sketch	type code	feature dimension code of expressing bearing fits and installation	Bearing basic code	standard number
needle roller bearing	open type drawn cup needle roller bearing		HK	$F_w B^{①}$	$HKF_w B$	GB/T 290
	close type drawn cup needle roller bearing		BK	$F_w B^{①}$	$BKF_w B$	

Note: in table F_w — needle roller set inner diameter of needle roller bearing without inner ring (needle roller and cage assembly inner diameter) ;

E_w — needle roller and cage assembly outer diameter; B — nominal bearing width; B_c — needle roller and cage assembly width; D_{el} — thrust needle roller and cage assembly inner ring diameter; D_e — thrust needle roller and cage assembly outer diameter.

- ① when expressing dimension directly by numbers of mm, if it is a single digit number, should add “0” on its left, such as 8 mm is express by 08.
 ② Except the diameter less than 10 mm expressed by “actual nominal numbers of mm”, all other inner diameter codes are espressed according to Table 1 - 6.

4. Prefix code

Bearing prefix code and its meaning are shown in Table 1 – 9.

Table 1 – 9 Prefix code

code	meaning	example
L	separable inner ring or outer ring of separable bearing	LNU 207 LN207
R	bearing without separable inner ring or outer ring (for needle roller bearing only suit able for NA type)	RUN 207 RNA 6904
K	roller and cage assembly	K 81107
WS	thrust cylindrical roller bearing shaft washer	WS 81107
GS	thrust cylindrical roller bearing housing washer	GS 81107
F	radial ball bearing with flanged outer ring (only for $d \leq 10\text{mm}$)	F618/4
KOW-	thrust bearing without shaft washer	KOW-51108
KIW-	thrust bearing without housing washer	KIW-51108
LR	bearing with separable inner ring or outer ring and rolling element set	—

5. Suffix code

1) Inner structure codes are shown in Table 1 – 10.

Table 1 – 10 Inner structure code

code	meaning	example
A	1) double row angular contact or deep groove ball bearing without filling slot 2) needle roller bearing outer ring with two retaining snap ring ($d > 9\text{mm}$, $F_w > 12\text{mm}$) 3) deep groove ball bearing with ring without rib	3205A — —
B	1) angular contact ball bearing with nominal contact angle $\alpha = 40^\circ$ 2) tapered roller bearing with larger contact angle	7210B 32310 B
C	1) angular contact ball bearing with nominal contact angle $\alpha = 15^\circ$ 2) aligning roller bearing with design alteration, inner ring without rib, movable center ring, pressed steel cage, symmetrical type roller, reinforcement type	7005C 23122C
CA	C type aligning roller bearing, inner ring with rib, movable center ring, machined cage	23084 CA/W33
CC	C type aligning roller bearing, roller guiding improvement Note: there is also a second explanation for CC., see Table 1 – 16.	22205 CC
CAB	CA type aligning roller bearing, through bore on roller center, cage with pin	—

(continued)

code	meaning	example
CABC	CAB type aligning roller bearing, roller guiding improvement	—
CAC	CA type aligning roller bearing, roller guiding improvement	22252 CACK
AC	angular contact ball bearing with nominal contact angle $\alpha = 25^\circ$	7210AC
D	split bearing	K50 * 55 * 20D
ZW	needle roller and cage assembly double row	K20 * 25 * 40ZW
E	reinforcement type, that is inner structure design improvement, increase loading capacity.	NU 207E

Note: in table d — needle roller bearing bore diameter; F_w — needle roller complement bore diameter of needle roller bearing without inner ring.

2) Sealed, shielded and outer side shape change codes are shown in Table 1 – 11.

Table 1 – 11 Sealed, shielded and outside shape change

code	meaning	example
K	tapered bore bearing cone 1:12 (except insert ball bearing)	1210K
K30	tapered bore bearing cone 1:30	24122 K30
R	bearing outer ring with snap rib (flange outer ring) (not suited for radial ball bearing of bore diameter smaller than 10mm)	30307R
N	bearing outer ring with snap ring groove	6210N
NR	bearing outer ring with snap ring groove, and with snap ring	6210NR
-RS	bearing with skeleton rubber seal on one side (contact type)	6210-RS
-2RS	bearing with skeleton rubber seals on two sides (contact type)	6210-2RS
-RZ	bearing on one side with skeleton rubber seal (non contact type)	6210-RZ
-2RZ	bearing with skeleton rubber seals on two sides (non contact type)	6210-2RZ
-Z	bearing with shield on one side	6210-Z
-2Z	bearing with shield on two side	6210-2Z
-RSZ	bearing with skeleton rubber seal on one side (contact type) with shield on one side	6210-RSZ
-RZZ	bearing with skeleton rubber seal on one side (non contact type) with shield on one side	6210-RZZ
-ZN	bearing with shield on one side, outer ring with snap ring groove on another side	6210-ZN
-ZNR	bearing with shield on one side, outer ring with snap ring groove and snap ring on another side	6210-ZNR
-ZNR	bearing with shield on one side, outer ring with snap ring groove on same side	6210-ZNB
-2ZN	bearing with shield on two side, outer ring with snap ring groove	6210-2ZN

(continued)

code	meaning	example
-U	thrust ball bearing with spherical surface washer	53210 U
-FS	bearing with felt seal on one side	6203-FS
-2FS	bearing with felt seal on two sides	6206-2FSWB
-LS	bearing with skeleton rubber seal on one side (contact type, no slot on ring)	—
-2LS	bearing with skeleton rubber seal on two sides (contact type, no slot on ring)	NNF 5012-2LSNV
	Note: -LS and -2LS are mostly used for roller bearing.	
PP	bearing with soft rubber seal on two sides	NATR 8 PP
-2K	double row tapered bore bearing cone 1:12	QF 2308-2K
D	1) double row angular contact ball bearing, double inner ring, contact angle $\alpha = 45^\circ$	3307 D
	2) double row tapered roller bearing without inner ring not decorate grinding end	—
DC	double row angular contact ball bearing, double outer ring	3924-2KDC
D1	double row tapered roller bearing without inner ring decorate grinding end	—
DH	single direction thrust bearing with two housing washers	—
DS	single direction thrust bearing with two shaft washers	—
N1	bearing outer ring with one location slot	—
N2	bearing outer ring with two or more than two symmetrical location slots	—
N4	N + N2 location slot and snap groove on different side	—
N6	N + N2 location slot and snap groove on same side	—
P	aligning roller bearing with two-piece outer ring	—
PR	same as P, between two-piece has a spacer ring	—
S	1) bearing outer ring outside surface is spherical surface (except insert bearing)	—
	2) clearance can be adjusted (needle roller bearing)	NA4906 S
WB	extended inner ring (double side extended); WB1-one side extended	—
WC	extended outer ring bearing	—
SC	radial bearing with outside cover	—
X	track needle roller bearing with outer ring outside surface as cylindrical surface	KR 30X
Z	needle roller combined bearing with shielded cover	NK 25 Z
ZH	thrust bearing, housing washer with shielded cover	—
ZS	thrust bearing, shaft washer with shielded cover	—

Note: seal code and shield code can combine with snap groove code.

3) Cage structure code and material change code is shown in Table 1 – 12.

Table 1 – 12 Cage structure code and material change code

Code	meaning
a) cage material	
F	steel、nodular cast iron or powder metallurgy machined cage, express by different number for different material
F1	carbon steel
F2	graphite
F3	nodular cast iron
F4	powder metallurgy
Q	machined bronze cage, express by additional number for different material
Q1	aluminum iron manganese bronze
Q2	silicon iron zinc bronze
Q3	silicon nickel bronze
Q4	aluminum bronze
M	machined brass cage
L	machined light alloy cage, express by additional number for different material
L1	LY11CZ hard aluminum 11
L2	LY12CZ hard aluminum 12
T	machined phenolic aidehyde lamination cloth tube cage
TH	glass fiber reinforced phnolic resin cage (basket type)
TN	engineering plastics molding cage, express by additional number for different material
TN1	nylon
TN2	polysuphpne
TN3	polyimide
TN4	polycarboate
TN5	polyfomaldehyde
J	steel plate press cage express by additional number for material change
Y	brass plate press cage express by additional number for material change
SZ	cage made by spring filament or spring
V	full complement bearing (without cage)
b) cage structure type and surface treatment	
H	self-locked pocket cage
W	welding cage
R	riveting cage (for large bearing)
E	phosphoric treatment cage

(continued)

Code	meaning
b) cage structure type and surface treatment	
D	carburizing and nitrogen cementation cage
D1	carburizing cementation cage
D2	nitrogen case hardening cage
C	plating layer cage (C1-silver-plating)
A	outer ring guiding
B	inner ring guiding
P	window cage with inner ring guiding or outer ring guiding and with broaching hole or pressing hole
S	guiding surface with lubricating slot

Note: 1. The code in “b” can only be used combining with the code in “a” altogether. such as MPS — machined brass cage (window cage) with broaching hole or pressing hole, outer ring or inner ring guiding, guiding surface with lubricating slot.
JA — steel plate pressed cage, outer ring guiding, FE — phosphoric treatment machined steel cage.
2. While bearing cage using the following structure and material, not need compile the code of cage material change.

ordinal number	bearing type	cage structure and material
1	deep groove ball bearing	1) while bearing outer diameter $D \leq 400\text{mm}$, using steel plate (band) or brass (band) pressed cage 2) while bearing outer diameter $D > 400\text{mm}$, using machined brass cage
2	aligning ball bearing	1) while bearing outer diameter $D \leq 200\text{mm}$, using steel plate (band) pressed cage 2) while bearing outer diameter $D > 200\text{mm}$, using machined brass cage
3	cylindrical roller bearing	1) cylindrical roller bearing: while bearing outer diameter $D \leq 400\text{ mm}$, using steel plate (band) pressed cage; while bearing outer diameter $D > 400\text{mm}$, using machined steel cage 2) For double row cylindrical roller bearing, using machined brass cage
4	aligning roller bearing	1) For convex symmetrical bearing (with movable guide ring) , using steel plate (band) pressed cage 2) For other aligning roller bearing, using machined brass cage
5	needle roller bearing long cylindrical roller bearing	using steel plate or hard aluminum pressed cage using steel plate (band) pressed cage

(continued)

ordinal number	bearing type	cage structure and material
6	angular contact ball bearing	1) For separable angular contact ball bearing, using machined phenolic aidehyde lamination cloth tube cage 2) For two-piece inner ring or outer ring (three-point, four-point contact) ball bearing, using machined aluminum cage 3) angular contact ball bearing and its variety type while bearing outer diameter $D \leq 250\text{mm}$, contact angle $\alpha = 15^\circ$ or $\alpha = 25^\circ$, using machined phenolic aidehyde lamination cloth tube cage; $\alpha = 40^\circ$, using steel plate pressed cage while bearing outer diameter $D > 250\text{mm}$, using machined brass or hard aluminum cage, for P5, P4, P2 grade bearing, using machined phenolic aidhyde lamination cloth tube cage for angular contact ball bearing with stepped inner ring and its variety, using machined phenolic aidhyde lamination cloth tube cage 4) For double row angular contact ball bearing, using steel plate (band) pressed cage
7	Tapered roller bearing	1) while bearing outer diameter $D \leq 650\text{mm}$, using steel plate pressed cage 2) while bearing outer diameter $D > 650\text{mm}$, using machined steel cage
8	thrust ball bearing	1) while bearing outer diameter $D \leq 250\text{mm}$, using steel plate (band) pressed cage 2) while bearing outer diameter $D > 250\text{mm}$, using machined cage
9	thrust roller bearing	1) For thrust cylindrical roller bearing, using machined cage 2) For thrust self-aligning roller bearing, using machined cage 3) For thrust tapered roller bearing, using machined cage 4) For thrust needle roller bearing, using pressed cage

4) Bearing parts material code is shown in Table 1 – 13.

Table 1 – 13 Bearing parts material change code

suffix code	meaning	example
/HE	ring and rolling element and cage or only ring and rolling element made by Electroslag Refining ZGCr 15	6204/HE
/HA	ring and rolling element and cage or only ring and rolling element made by Vacuum Induction Melting Bearing Steel	6204/HA
/HU	ring and rolling element and cage or only ring and rolling element made by non-tempered stainless steel 1Cr18Ni9Ti	6004/HU
/HV	ring and rolling element and cage or only ring and rolling element made by tempered stainless steel (/HV — 9Gr18; /HV1 — 9Cr18Mo)	6014/HV

(continued)

suffix code	meaning	example
/HN	ring and rolling element made by heat-resisting steel (/HN — Cr4Mo4V ; /HN1 — Cr14Mo4 ; /HN2 — Cr15Mo4V ; /HN3 — W18Cr4V)	NU 208/HN
/HC	ring and rolling element and cage or only ring made by carburizing steel (/HC — 20Cr2Ni4A ; /HC120Cr2Mn2MoA ; /HC2 — 15Mn)	—
/HP	ring and rolling element made by bronze and other anti-magnetic material, while material changing, express by additional number	—
/HQ	ring and rolling element made by non-commonly used material (/HQ-plastic ; /HQ1-ceramics)	—
/HG	ring and rolling element or only ring made by other bearing steel (/HG — 5CrMnMo ; /HG1 — 55SiMoVA)	—

Note: ordinary bearing parts are made by GCr15 bearing steel, which code is omitted.

5) Tolerance grade code is shown in Table 1 – 14.

Table 1 – 14 Tolerance grade code

code	meaning	example
/P0	tolerance grade conforms to the standard of 0 grade, omit in code, not expressed	6203
/P6	tolerance grade conforms to the standard of 6 grade	6203/P6
/P6x	tolerance grade conforms to the standard of 6x grade	30210/P6x
/P5	tolerance grade conforms to the standard of 5 grade	6203/P5
/P4	tolerance grade conforms to the standard of 4 grade	6203/P4
/P2	tolerance grade conforms to the standard of 2 grade	6203/P2
/SP	dimension tolerance equivalent to P5 grade, running accuracy equivalent toP4	234420/SP
/UP	dimension tolerance equivalent to P4 grade, running accuracy equivalent toP4	234730/UP

6) Bearing clearance code is shown in Table 1 – 15.

Table 1 – 15 Bearing clearance code

code	meaning	example
/C1	clearance conforms to the standard of 1 group	NN 3006K/C1
/C2	clearance conforms to the standard of 2 group	6210/C2
—	clearance conforms to the standard of 0 group	6210
/C3	clearance conforms to the standard of 3 group	6210/C3
/C4	clearance conforms to the standard of 4 group	NN 3006K/C4
/C5	clearance conforms to the standard of 5 group	NNU 4920K/C5

(continued)

code	meaning	example
/CN	0 group clearance. /CN combined with letter H, M or L express clearance range reducing to half, or combined with P, expresses clearance range shifting to one side. Such as: /CNH — reducing to half of 0 group clearance range, located on upper half part /CNM — reducing to half of 0 group clearance range, located on middle part /CNL — reducing to half of 0 group clearance range, located on lower half part	—
/C9	/CNP — clearance range located on 0 group upper half part and on C3 group lower half part; bearing clearance is different from the present standard.	6205-2RS/C9

Note: While expressing tolerance code and clearance code at the same time, simplification can be made. That is to combine tolerance grade code adding clearance group number (0 group is not expressed).

Such as: /P63 — expressing tolerance grade as P6 grade, radial clearance as 3 group
/P52 — expressing tolerance grade as P5 grade, radial clearance as 2 group

7) Bearing arrangement code is shown in Table 1 – 16.

Table 1 – 16 Bearing arrangement code

code	meaning	example
/DB	paired mounting of back-to-back arrangement	7210C/DB
/DF	paired mounting of face-to-face arrangement	32208/DF
/DT	paired mounting of tandem arrangement	7210C/DT
a) bearing number in arrangement		
/D	two bearings	
/T	three bearings	
/Q	four bearings	
/P	five bearings	
/S	six bearings	
b) bearing arrangement in arrangement group		
B	back to back	
F	face to face	
T	tandem	
BT	back to back and tandem	
FT	face to face and tandem	
BC	back to back of paired tandem	

(continued)

code	meaning	example
FC	face to face of paired tandem	
Note: many arrangement forms can be composed of combining a and b: such as, paired mounting of arrangement DB, DF, DT, three bearing arrangement TBT, TTFT, TT and four bearing arrangement QBC, QFC, QT, QBT, QFT and so on.		
c) adding letter after arrangement code means to have the meaning of		
GA	light preload (deep groove and angular contact ball bearing)	} if using in angular contact ball bearing, omit “G”
GB	medium preload (deep groove and angular contact ball bearing)	
GC	heavy preload (deep groove and angular contact ball bearing)	
G	special preload, additional number expresses preload value	
CA	smaller axial internal clearance (deep groove and angular contact ball bearing)	}
CB	axial internal clearance larger than CA (deep groove and angular contact ball bearing)	
CC	axial internal clearance larger than CB (deep groove and angular contact ball bearing)	
CG	axial internal clearance is zero (tapered roller bearing)	
R	radial load even distribution	

8) Other special characteristic code is shown in Table1 – 17.

Table 1 – 17 other special characteristic code

code	meaning	example
/Z	bearing vibration acceleration grade; additional number expresses different limit value Z1 — bearing vibration acceleration grade limit value conforms to the standard value of Z1 group Z2 — bearing vibration acceleration grade limit value conforms to the standard value of Z2 group Z3 — bearing vibration acceleration grade limit value conforms to the standard value of Z3 group	6204/Z1 6205-2RS/Z2
/V	bearing vibration velocity grade limit value. additional number expresses different limit value V1 — bearing vibration velocity limit value conforms to the standard value of V1 group V2 — bearing vibration velocity limit value conforms to the standard value of V2 group V3 — bearing vibration velocity limit value conforms to the standard value of V3 group	6306/V1 6304/V2
/ZC	stipulating value for bearing noise grade. Additional number expresses different limit value	—

(continued)

code	meaning	example
/T	having requirement for bearing in starting toque , afterward number expresses starting toque	—
/RT	having requirement for bearing in running toque , afterward number expresses running torque	—
/S0	bearing ring experiences high temperature tempering treatment , working temperature can be 150℃	N210/S0
/S1	bearing ring experiences high temperature tempering treatment , working temperature can be 200℃	NUP 212/S1
/S2	bearing ring experiences high temperature tempering treatment , working temperature can be 250℃	NU 214/S2
/S3	bearing ring experiences high temperature tempering treatment , working temperature can be 300℃	NU 308/S3
/S4	bearing ring experiences high temperature tempering treatment , working temperature can be 350℃	NU 214/S4
/W20	bearing outer ring with three lubrication oil holes	—
/W26	bearing inner ring with six lubrication oil holes	—
/W33	bearing outer ring with lubrication groove and three lubrication oil holes	23120CC/W33
/W33X	bearing outer ring with lubrication groove and six lubrication oil holes	—
/W513	W26 + W33	—
W518	W20 + W26	—
/AS	outer ring with oil hole , additional number expresses the number of oil hole (needle roller bearing)	HK 2020/AS1
/IS	inner ring with oil hole , additional number expresses the number of oil hole (needle roller bearing)	NAO
	after AS and IS adding “R” respectively expresses inner ring	17 * 30 * 13/IS1
	or outer ring with lubrication oil hole and groove	NAO15 * 28 * 13/ASR
/HT	injecting special high temperature grease into bearing ; while the injecting grease quantity in bearing is different from the standard quantity , expresses it by additional letter: A — injecting grease quantity less than the standard quantity B — injecting grease quantity more than the standard quantity C — injecting grease quantity more than B (full filling)	NA 6909/ISR/HT
/LT	injecting special low temperature grease. Additional letter means as same as HT.	—
	injecting special medium temperature grease. Additional letter means as same as HT.	—

(continued)

code	meaning	example
/LHT	injecting special high, low temperature grease. Additional letter means as same as HT.	—
/Y	combining Y and another letter (such as YA、YB) or also adding number can be used for distinguishing non-series changing that can not be expressed by existing suffix code. YA — structure change (synthesis expression) YA1 — outside surface of bearing outer ring has discrepancy with standard design YA2 — bore of bearing inner ring has discrepancy with standard design YA3 — face of bearing ring has discrepancy with standard design YA4 — raceway of bearing ring has discrepancy with standard design YA5 — rolling element of bearing has discrepancy with standard design	—
/Y	YB — technical factor change (synthesis expression) YB1 — having plating layer on bearing ring surface YB2 — changing for bearing dimension and tolerance requirement YB3 — changing for bearing ring surface roughness requirement YB4 — changing for heat treatment requirement (such as hardness) Note: All of having Y and another letter or adding number in suffix code in bearing code, then for understanding the the specific changing content, must check the drawing or the supplementary technical condition	

1.3.2 Identification Codes for Bearings with Subunit

Identification codes for bearings with subunit is shown in Table 1 – 18.

Table 1 – 18 Identification codes for bearings with subunit

Subunit name	bearing with subunit code ^①	example
with adapter sleeve	bearing code + adapter code	22208K + H308
with withdrawal sleeve	bearing code + withdrawal code	22208K + AH308

(continued)

Subunit name	bearing with subunit code ^①	example
with inner ring	bearing code + IR suitable for needle roller bearing without inner ring, needle roller combined bearing	HKX 30 + IR
with thrust collar	bearing code + thrust collar code ^② suitable for cylindrical roller bearing	NJ 210 + HJ210

- ① Only suitable for the mark on the package of the bearing with subunit and the drawings, design files, handbook. Not suit able for bearing mark.
- ② Can combine and simplify NJ... + HJ... = NH, such as: NH 210.

1.4 Rolling Bearing Accuracy and Inspection

1.4.1 Tolerance Grades of Rolling Bearings

According to dimension tolerance and running accuracy , rolling bearing tolerance can be divided into five grades from high to low , whose value is according to the GB/T 307. 1 and GB/T 307. 4.

Radial bearings (except tapered roller bearings) can be divided into 0 , 6 , 5 , 4 , 2 grades (corresponding to ISO 0 , 6 , 5 , 4 and 2 grades respectively) .

1.4.2 Symbols and Definitions

1. Outside dimension and running accuracy

Figure 1 – 32 shows the schematic outside dimensions of a rolling bearing. The designatoin of dimension tolerance and running accuracy are shown in Table 1 – 19.

Table 1 – 19 The designations of dimension tolerance and running accuracy of rolling bearings

symbol	meaning
B	inner ring width
V_{Bs}	variation of inner ring width
Δ_{Bs}	deviation of a single inner ring width
C	outer ring width
C_1	outer ring flange width

(continued)

symbol	meaning
V_{Cs}	variation of outer ring width
V_{C1s}	variation of outer ring flange width
Δ_{Cs}	deviation of a single outer ring width
Δ_{C1s}	deviation of a single outer ring flange width
d	nominal bore diameter
d_1	theoretical large end diameter of basic tapered bore
V_{dmp}	variation of mean bore diameter (only suitable for basic cylindrical bore)
V_{dsp}	variation of bore diameter in a single plane
Δ_{dmp}	deviation of mean bore diameter in a single plane (for basic tapered bore, only is for theoretical small end bore diameter)
Δ_{ds}	deviation of a single bore diameter
Δ_{dlmp}	deviation of mean bore diameter in a single plane of theoretical large end diameter of basic tapered bore
D	outside diameter
D_1	outside diameter of outer ring flange
V_{Dmp}	variation of mean outside diameter
V_{DSP}	variation of outside diameter in a single plane
Δ_{Dmp}	deviation of mean outside diameter in a single plane
Δ_{Ds}	deviation of outside diameter in a single plane
Δ_{D1s}	deviation of outer ring flange outside diameter in a single plane
K_{ea}	radial runout of outer ring of assembled bearing
K_{ia}	radial runout of inner ring of assembled bearing
S_d	axial runout of inner ring face with respect to the bore
S_D	axial runout of outer ring outside surface with respect to the face
S_{D1}	axial runout of outer ring outside surface with respect to the flange back face
S_{ea}	axial runout of assembled bearing outside ring
S_{eal}	axial runout of outer ring flange back face of assembled bearing
S_{ia}	axial runout of inner ring of assembled bearing
α	tapered angle of inner ring bore (half tapered angle)

2. Additional symbols of tapered roller bearings

The additional symbols of tapered roller bearings are shown in Figure 1 – 32.

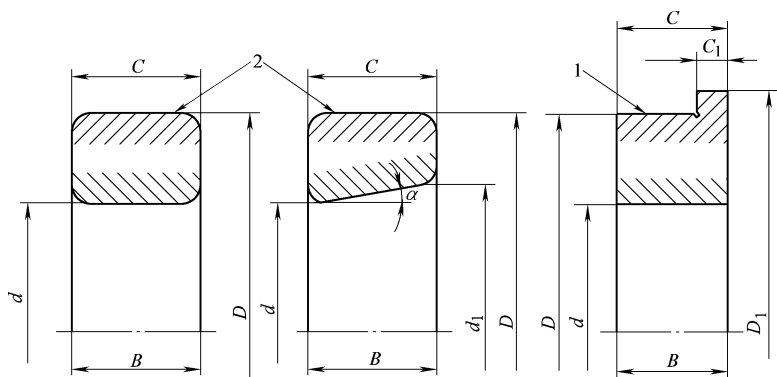


Figure 1-32 Outside dimension designations

T — width of assembled bearing

T_1 — effective width of inner assembly

T_2 — effective width of outer ring

and let

Δ_{Ts} — actual width deviation of assembled bearing

Δ_{T1s} — actual effective width deviation of inner assembly

Δ_{T2s} — actual effective width deviation of outer ring

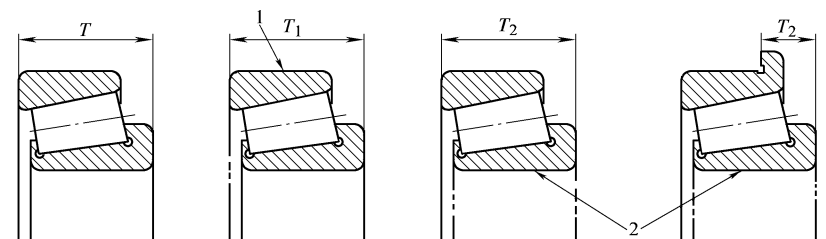


Figure 1-33 Additional symbols for tapered roller bearings

1 — standard outer ring 2 — standard inner assembly

3. Vocabulary associated with tolerances

(1) Bore diameter and outside diameter

1) Nominal bore diameter [outside diameter]

A diameter of the cylinder containing the theoretical surface of a basically cylindrical bore [cylindrical outside surface], or diameter, in a designated radial plane, of the cone containing the theoretical surface of a

basically tapered bore, or diameter of the sphere containing the theoretical surface of a basically spherical outside surface Note-For rolling bearings, the nominal bore and outside diameter are generally the reference values (basic diameters for the deviations of the actual bore and outside surface)

2) Single bore diameter [outside diameter]

A distance between two parallels tangential to the line of intersection of the actual bore surface [outside surface] and a radial plane.

3) Deviation of a single bore diameter [outside diameter]

A difference between a single bore diameter [outside diameter] and the nominal bore diameter [outside diameter] of a basically cylindrical bore [outside surface]

4) Variation of bore diameter [outside diameter]

A difference between the largest and the smallest single bore diameters [single outside diameters] of an individual ring or washer with a basically cylindrical bore [outside surface].

5) Mean bore diameter [outside diameter]

An arithmetical mean of the largest and smallest single bore diameters [single outside diameters] of an individual ring or washer with a basically cylindrical bore [outside surface].

6) Deviation of mean bore diameter [outside diameter]

A difference between the mean bore diameter [mean outside diameter] and the nominal bore diameter of a basically cylindrical bore [outside surface] in a single radial plane.

7) Mean bore diameter [outside diameter] in a single plane

An arithmetical mean of the largest and smallest single bore diameters [single outside diameters] in a single radial plane.

8) Deviation of mean bore diameter [outside diameter] in a single plane

A difference between the mean bore diameter [mean outside diameter] and the nominal bore diameter [nominal outside diameter] of a basically cylindrical bore [outside surface] in a single radial plane.

9) Variation of a single bore diameter [outside diameter] in a single

plane

A difference between the largest and the smallest single bore diameters [single outside diameters] in a single plane.

10) Variation of mean bore diameter [outside diameter]

A difference between the largest and smallest single plane mean bore diameters [mean outside diameters] of an individual ring or washer with a basically cylindrical bore [outside surface].

(2) Width and height

1) Nominal ring width

A distance between the two theoretical side face of a ring. Note-For bearing rings, the nominal width is generally the reference value (basic dimension) for deviation of the actual width).

2) Single ring width

A distance between the points of intersection of the two actual side faces of a ring and a straight line perpendicular to the plane tangential to the reference face of the ring.

3) Deviation of a single ring width

A difference between a single ring width and the nominal ring width.

4) Variation of ring width

A difference between the largest and the smallest single ring widths of an individual ring.

5) Mean ring width

An arithmetical mean of the largest and smallest single ring widths of an individual ring.

6) Nominal bearing width [bearing height]

A distance between the two theoretical ring faces [washer back faces] designated to bound the width of a radial bearing [the height of a thrust bearing].

Note: The nominal bearing width or nominal bearing height is generally the reference value (basic dimension) for deviations of the actual bearing width or actual bearing height

7) Actual bearing width

A distance between the points of intersection of the bearing axis and the two planes tangential to the actual ring faces designated to bound the width

of a radial bearing where one inner ring face and one outer ring face are designated to bound the width.

Note: For a single-row tapered roller bearing, this is the distance between the points of intersection of the bearing axis and two planes, one of which is tangential to the actual back face of the inner ring and one of which is tangential to that of the outer ring, the inner ring and outer ring raceway and the inner ring back race rib being in contact with all the rollers:

8) Deviation of the actual bearing width

A difference between the actual bearing width and the nominal bearing width of a radial bearing.

9) Actual bearing height

A distance between the points of intersection of the bearing axis and the two planes tangential to the actual washer back faces designated to bound the height of a thrust bearing.

10) Deviation of the actual bearing height

A difference between the actual bearing height and the nominal bearing height of a thrust bearing.

(3) Chamfer dimensions

1) Nominal chamfer dimension

A ring chamfer dimension value for reference purposes.

Note: The nominal chamfer dimension corresponds to the smallest single chamfer dimension.

2) Radial single chamfer dimension

A distance, in a single axial plane, between the imaginary sharp corner of a ring or washer and the intersection of a chamfer surface and the face of the ring or washer.

3) Axial single chamfer dimension

A distance, in a single axial plane, between the imaginary sharp corner of a ring or washer and the intersection of a chamfer surface and the bore or outer surface of the ring or washer

4) Smallest single chamfer dimension

The smallest permissible radial and axial single chamfer dimensions of a ring or washer; in addition, the radius of an imaginary circular arc, in an axial plane, tangential to the ring or washer face and to the bore or outer surface of the ring or washer, beyond which no ring or washer material is

allowed.

5) Largest single chamfer dimension

The largest permissible radial and axial single chamfer dimension.

(4) Ball dimension

1) Nominal ball diameter

A diameter value which is used for the general identification of a ball size.

2) Single ball diameter

A distance between two parallel planes tangential to the actual surfaces of a ball.

3) Mean ball diameter

An arithmetical mean of the largest and the smallest single diameter of a ball.

4) Variation of ball diameter

A difference between the largest and the smallest single diameters of a ball.

5) Ball lot

A defined number of balls manufactured under the same conditions and which are considered as an entity.

6) Mean diameter of (ball) lot

An arithmetical mean of the mean diameters of the largest ball and the smallest ball in a ball lot.

7) Variation of (ball) lot diameter

A difference between the mean diameters of the largest ball and the smallest ball in a ball lot.

8) Ball grade

A specific combination of dimension, shape, surface roughness and group tolerance for balls.

9) Ball gauge

An amount by which the mean diameter of a ball lot should differ from the nominal ball diameter, this amount being one of an established series.

10) Deviation of a ball lot from ball gauge

A difference between the mean diameter of a ball lot and the sum of the

nominal ball diameter and the ball gauge.

11) Ball subgauge

An amount, of an established series of amounts, is the nearest to the actual deviation from ball gauge of a ball lot.

(5) Roller dimensions

1) Nominal roller diameter

A diameter value used for the general identification of roller diameter.

Note: For a symmetrical roller this is the theoretical diameter in a radial plane through the middle of the roller length and for an asymmetrical roller it is the largest theoretical diameter (i. e. in a radial plane at the imaginary sharp large end corner of a tapered roller).

2) Single roller diameter

A distance between two tangents to the actual surface of a roller parallel to each other and in a plane perpendicular to the roller axis (a radial plane).

3) Mean roller diameter in a single plane

An arithmetical mean of the largest and smallest single roller diameters in a single radial plane.

4) Variation of roller diameter in a single plane

A difference between the largest and smallest single roller diameters in a single radial plane.

5) Nominal roller length

A length value used for the general identification of roller length.

6) Actual roller length

A distance between the two radial planes which just contain the extremities of a roller.

7) Roller gauge

A diameter deviation range limited by a high and a low deviation of the mean roller diameter in a single plane from the nominal roller diameter in the same specified radial plane.

Note: For cylindrical rollers and needle rollers, the plane through the middle of the roller length is used.

8) (roller) Gauge lot

A quantity of rollers, of the same roller grade and nominal dimensions,

all having a mean roller diameter in a single plane within the same roller gauge.

9) Variation of (roller) gauge lot diameter

A difference between the mean diameter in a single plane of the roller having the largest such diameter and that of the roller having the smallest such diameter in a roller gauge lot.

10) Roller grade

A specific combination of dimension, shape surface roughness and group tolerances for rollers

(6) Shape

1) Deviation from circular shape (of basically circular line on a surface)

The greatest radial distance between the circle inscribed in the line (inside surface) or circumscribed around the line (outside surface) and any point on the line.

2) Deviation from cylindrical shape (of basically cylindrical surface)

The greatest radial distance in a radial plane, between the cylinder inscribed in the surface (inside surface) or circumscribed around the surface (outside surface) and any point on the surface.

3) Deviation from spherical shape (of basically spherical surface)

The greatest radial distance in any equatorial plane, between the sphere inscribed in the surface (inside surface) or circumscribed around the surface (outside surface) and any point on the surface.

(7) Running accuracy

1) Radial runout of inner ring of assembled bearing (radial bearing)

A difference between the largest and the smallest of the radial distances between the bore surface of the inner ring, in different angular positions of this ring, and a point in a fixed position relative to the outer ring.

Note: At the angular position of the point mentioned, or on each side and close to it, rolling elements should be in contact with both the inner ring and outer ring raceways and (in a tapered roller bearing) the inner ring rib.

2) Radial runout of outer ring of assembled bearing (radial bearing)

A difference between the largest and the smallest radial distances

between the outside surface of the outer ring, in different angular positions of this ring, and a point in a fixed position relative to the inner ring.

Note: At the angular position of the point mentioned, or on each side and close to it, rolling elements should be in contact with both the inner and outer ring raceways and (in a tapered roller bearing) the inner ring back face rib.

3) Axial runout of inner ring of assembled bearing (radial groove ball bearing)

A difference between the largest and the smallest axial distances between the reference face of the inner ring, in different angular positions of this ring, at a radial distance from the inner ring axis equal to half of the inner ring raceway contact diameter, and a point in a fixed position relative to the outer ring.

Note: the inner ring and outer ring raceways should be in contact with all the balls.

4) Axial runout of inner ring of assembled bearing (tapered roller bearing)

A difference between the largest and the smallest axial distances between the back face of the inner ring, in different angular positions of this ring, at a radial distance from the inner ring axis equal to half of the inner ring mean raceway contact diameter, and a point in a fixed position relative to the outer ring.

Note: The inner ring and outer ring raceways and the inner ring back face rib should be in contact with all the rollers.

5) Axial runout of outer ring of assembled bearing (radial groove ball bearing)

A difference between the largest and smallest axial distances between the reference face of the outer ring, in different angular positions of this ring, at a radial distance from the outer ring axis equal to half of the outer ring raceway contact diameter, and a point in a fixed position relative to the inner ring.

Note: The inner and outer ring raceways should be in contact with all the balls.

6) Axial runout of outer ring of assembled bearing (tapered roller bearing)

A difference between the largest and the smallest axial distances

between the back face of the outer ring, in different angular positions of this ring, at a radial distance from the outer ring axis equal to half of the outer ring mean raceway contact diameter, and a point in a fixed position relative to the inner ring.

Note: The inner ring and outer ring raceways and the inner ring back face rib should be in contact with all the rollers.

7) Axial runout of inner ring face with respect to the bore

A difference between the largest and smallest axial distances between a plane perpendicular to the inner ring axis and the reference face of the inner ring, at a radial distance from the inner ring axis equal to half of the mean diameter of the face.

8) Parallelism of raceway with respect to the face (inner or outer ring of radial groove ball bearing)

A difference between the largest and the smallest axial distances between the plane tangential to the reference face and the plane of the middle of the raceway.

9) Variation of outer ring outside surface generatrix inclination with respect to the face (of a basically cylindrical surface)

A total variation of relative positions, in a radial direction parallel with the plane tangential to the reference face of the outer ring, of points on the same generatrix of the outside surface at a distance from the side faces of the outer ring equal to the largest axial single chamfer dimension.

10) Variation in thickness between inner ring raceway and bore surface (radial bearing)

A difference between the largest and smallest radial distance between the bore surface and the middle plane of the raceway on the inner ring.

11) Variation in thickness between outer ring raceway and outside surface (radial bearing)

A difference between the largest and the smallest radial distances between the outside surface and the middle plane of the raceway on the outer ring.

12) Variation in thickness between washer raceway and back face

(thrust bearing shaft or housing washer, flat back face)

A difference between the largest and smallest axial distances between the back face and the middle plane of the raceway on the possible side of the washer.

(8) Internal clearance

1) Radial internal clearance (bearing capable of taking purely radial load, non-preloaded)

An arithmetical mean of the radial distance through which one of the rings may be displaced relative to the other, from one eccentric extreme position to the diametrically opposite extreme position, in different angular directions and without being subjected to any external load.

Note: The mean value includes displacements with the rings in different angular positions relative to each other and with the set of rolling elements in different angular positions in relation to the rings.

2) Theoretical radial internal clearance (radial bearing)

An outer ring raceway contact diameter minus the inner ring raceway contact diameter minus twice the rolling element diameter.

3) Axial internal clearance (bearing capable of taking axial load in both directions, non-preloaded)

An arithmetical mean of the axial distances through which one of the rings or washers may be displaced relative to the other, from one axial extreme position to the opposite extreme position, without being subjected to any external load.

Note: The mean value include displacements with the rings or washers in different angular positions relative to each other and with the set of rolling elements in different angular positions in relation to the rings or washers.

1.4.3 Tables of Rolling Bearing Tolerance and Accuracy

1. Radial bearings (except tapered roller bearings)

Bore diameter tolerances given in Table 1 – 20 ~ Table 1 – 29 apply to basically cylindrical bores.

(1) Normal tolerance is given in Table 1 – 20 and Table 1 – 21.

Table 1-20 Inner ring

d/mm		$\Delta_{\text{dmp}}/\mu\text{m}$		$V_{\text{dsp}}/\mu\text{m}$			$V_{\text{dmp}}/\mu\text{m}$	$K_{\text{ia}}/\mu\text{m}$	$\Delta_{\text{Bs}}/\mu\text{m}$			$V_{\text{Bs}}/\mu\text{m}$
				diameter series					all	normal	modified ^①	
over	incl.	upper	lower	max			max	max	upper	lower		max
—	0.6	0	−8	10	8	6	6	10	0	−40	—	12
0.6	2.5	0	−8	10	8	6	6	10	0	−40	—	12
2.5	10	0	−8	10	8	6	6	10	0	−120	−250	15
10	18	0	−8	10	8	6	6	10	0	−120	−250	20
18	30	0	−10	13	10	8	8	13	0	−120	−250	20
30	50	0	−12	15	12	9	9	15	0	−120	−250	20
50	80	0	−15	19	19	11	11	20	0	−150	−380	25
80	120	0	−20	25	25	15	15	25	0	−200	−380	25
120	180	0	−25	31	31	19	19	30	0	−250	−500	30
180	250	0	−30	38	38	23	23	40	0	−300	−500	30
250	315	0	−35	44	44	26	26	50	0	−350	−500	35
315	400	0	−40	50	50	30	30	60	0	−400	−630	40
400	500	0	−45	56	56	34	34	65	0	−450	—	50
500	630	0	−50	63	63	38	38	70	0	−500	—	60
630	800	0	−75	—	—	—	—	80	0	−750	—	70
800	1000	0	−100	—	—	—	—	90	0	−1000	—	80
1000	1250	0	−125	—	—	—	—	100	0	−1250	—	100
1250	1600	0	−160	—	—	—	—	120	0	−1600	—	120
1600	2000	0	−200	—	—	—	—	140	0	−2000	—	140

① This applies to the inner rings and the outer rings of single bearings made for paired or stack mounting and also applies to the inner rings of $d \geq 50$ mm tapered bore bearings.

Table 1 – 21 Outer ring

D/mm		Δ_{Dmp} / μm		$V_{Dsp}^{①}/\mu m$				$V_{Dmp}^{①}$ / μm	K_{ea} / μm	Δ_{Cs} $\Delta_{Cls}^{②}$ / μm		V_{Cs} $V_{Cls}^{②}$ / μm	
				Open bearings		Capped bearings							
				diameter series									
				9	0、1	2、3、4	2、3、4						
over	incl.	upper	lower	max				max	max	upper	lower	max	
—	2.5	0	−8	10	8	6	10	6	15	identical to Δ_{Bs} and V_{Bs} of the inner ring of the same bearing			
2.5	6	0	−8	10	8	6	10	6	15				
6	18	0	−8	10	8	6	10	6	15				
18	30	0	−9	12	9	7	12	7	15				
30	50	0	−11	14	11	8	16	8	20				
50	80	0	−13	16	13	10	20	10	25				
80	120	0	−15	19	19	11	26	11	35				
120	150	0	−18	23	23	14	30	14	40				
150	180	0	−25	31	31	19	38	19	45				
180	250	0	−30	38	38	23	—	23	50				
250	315	0	−35	44	44	26	—	26	60				
315	400	0	−40	50	50	30	—	30	70				
400	500	0	−45	56	56	34	—	34	80				
500	630	0	−50	63	63	38	—	38	100				
630	800	0	−75	94	94	55	—	55	120				
800	1000	0	−100	125	125	75	—	75	140				
1000	1250	0	−125	—	—	—	—	—	160				
1250	1600	0	−160	—	—	—	—	—	190				
1600	2000	0	−200	—	—	—	—	—	220				
2000	2500	0	−250	—	—	—	—	—	250				

Note: The tolerances of outer ring flange outer diameters are given in Table 1 – 43.

① Applicable before mounting and after removal of internal or external snap rings.

② Only applies to groove ball bearings.

(2) Tolerance class 6 is given in Table 1 – 22 and Table 1 – 23.

Table 1 – 22 Inner ring

d/mm		Δ_{dmp} / μm		$V_{\text{dsp}}/\mu\text{m}$			V_{dmp} / μm	K_{ia} / μm	$\Delta_{\text{Bs}}/\mu\text{m}$			V_{Bs} / μm
				diameter series					all	normal	modified ^①	
over	incl.	upper	lower	max			max	max	upper	lower		max
—	0.6	0	−7	9	7	5	5	5	0	−40	—	12
0.6	2.5	0	−7	9	7	5	5	5	0	−40	—	12
2.5	10	0	−7	9	7	5	5	6	0	−120	−250	15
10	18	0	−7	9	7	5	5	7	0	−120	−250	20
18	30	0	−8	10	8	6	6	8	0	−120	−250	20
30	50	0	−10	13	10	8	8	10	0	−120	−250	20
50	80	0	−12	15	15	9	9	10	0	−150	−380	25
80	120	0	−15	19	19	11	11	13	0	−200	−380	25
120	180	0	−18	23	23	14	14	18	0	−250	−500	30
180	250	0	−22	28	28	17	17	20	0	−300	−500	30
250	315	0	−25	31	31	19	19	25	0	−350	−500	35
315	400	0	−30	38	38	23	23	30	0	−400	−630	40
400	500	0	−35	44	44	26	26	35	0	−450	—	45
500	630	0	−40	50	50	30	30	40	0	−500	—	50

① This applies to inner rings and outer rings of single bearings made for paired or stack mounting and also applies to inner rings of $d \geq 50$ mm tapered bore bearings.

Table 1-23 Outer ring

D/mm		Δ_{Dmp} / μm		$V_{\text{Dsp}}^{\text{①}}/\mu\text{m}$				$V_{\text{Dmp}}^{\text{①}}$ / μm	K_{ea} / μm	Δ_{Cs} $\Delta_{\text{Cls}}^{\text{②}}$ / μm		V_{Cs} $V_{\text{Cls}}^{\text{②}}$ / μm
				Open bearings		Capped bearings				upper	lower	max
				diameter series								
				9	0、1	2、3、4	0、1、2、3、4					
over	incl.	upper	lower	max				max	max	upper	lower	max
—	2.5	0	−7	9	7	5	9	5	8	identical to Δ_{Bs} and V_{Bs} of the inner ring of the same bearing		
2.5	6	0	−7	9	7	5	9	5	8			
6	18	0	−7	9	7	5	9	5	8			
18	30	0	−8	10	8	6	10	6	9			
30	50	0	−9	11	9	7	13	7	10			
50	80	0	−11	14	11	8	16	8	13			
80	120	0	−13	16	16	10	20	10	18			
120	150	0	−15	19	19	11	25	11	20			
150	180	0	−18	23	23	14	30	14	23			
180	250	0	−20	25	25	15	—	15	25			
250	315	0	−25	31	31	19	—	19	30			
315	400	0	−28	35	35	21	—	21	35			
400	500	0	−33	41	41	25	—	25	40			
500	630	0	−38	48	48	29	—	29	50			
630	800	0	−45	56	56	34	—	34	60			
800	1000	0	−60	75	75	45	—	45	75			

Note: The tolerances of outer ring flange outer diameters are given in Table 1-43.

① Applicable before mounting and after removal of internal or external snap ring.

② Only applies to groove ball bearings.

(3) Tolerance class 5 is given in Table 1 – 24 and Table 1 – 25.

Table 1 – 24 Inner ring

d/mm		$\Delta_{\text{dmp}}/\mu\text{m}$		$V_{\text{dsp}}/\mu\text{m}$		$V_{\text{dmp}}/\mu\text{m}$	$K_{\text{ia}}/\mu\text{m}$	$S_{\text{d}}/\mu\text{m}$	$S_{\text{ia}}^{\text{①}}/\mu\text{m}$	$\Delta_{\text{Bs}}/\mu\text{m}$			$V_{\text{Bs}}/\mu\text{m}$
				diameter series						all	normal	modified ^②	
				9	0、1 2、3、4								
over	incl.	upper	lower	max		max	max	max	max	upper	lower		max
—	0.6	0	−5	5	4	3	4	7	7	0	−40	−250	5
0.6	2.5	0	−5	5	4	3	4	7	7	0	−40	−250	5
2.5	10	0	−5	5	4	3	4	7	7	0	−40	−250	5
10	18	0	−5	5	4	3	4	7	7	0	−80	−250	5
18	30	0	−6	6	5	3	4	8	8	0	−120	−250	5
30	50	0	−8	8	6	4	5	8	8	0	−120	−250	5
50	80	0	−9	9	7	5	5	8	8	0	−150	−250	6
80	120	0	−10	10	8	5	6	9	9	0	−200	−380	7
120	180	0	−13	13	10	7	8	10	10	0	−250	−380	8
180	250	0	−15	15	12	8	10	11	13	0	−300	−500	10
250	315	0	−18	18	14	9	13	13	15	0	−350	−500	13
315	400	0	−23	23	18	12	15	15	20	0	−400	−630	15

① Only applies to groove ball bearings.

② This applies to inner rings and outer rings of single bearings made for paired or stack mounting and also applies to inner rings of $d \geq 50$ mm tapered bore bearings.

Table 1 - 25 Outer ring

D/mm		Δ_{Dmp} / μm		$V_{\text{Dsp}}/\mu\text{m}$		V_{Dmp} / μm	K_{ea} / μm	$S_{\text{D}}^{\text{①}}$ $S_{\text{Dl}}^{\text{②}}$ / μm	$S_{\text{ea}}^{\text{①②}}$ / μm	$S_{\text{eal}}^{\text{②}}$ / μm	Δ_{Cs} $\Delta_{\text{Cls}}^{\text{②}}$ / μm		V_{Cs} $V_{\text{Cls}}^{\text{②}}$ / μm
				diameter series									
				9	0、1、 2、3、4								
over	incl.	upper	lower	max		max	max	max	max	max	upper	lower	max
—	2.5	0	−5	5	4	3	5	8	8	11	Identical to Δ_{Bs} of the inner ring of the same bearing		5
2.5	6	0	−5	5	4	3	5	8	8	11			5
6	18	0	−5	5	4	3	5	8	8	11			5
18	30	0	−6	6	5	3	6	8	8	11			5
30	50	0	−7	7	5	4	7	8	8	11			5
50	80	0	−9	9	7	5	8	8	10	14			6
80	120	0	−10	10	8	5	10	9	11	16			8
120	150	0	−11	11	8	6	11	10	13	18			8
150	180	0	−13	13	10	7	13	10	14	20			8
180	250	0	−15	15	11	8	15	11	15	21			10
250	315	0	−18	18	14	9	18	13	18	25			11
315	400	0	−20	20	15	10	20	13	20	28			13
400	500	0	−23	23	17	12	23	15	23	33			15
500	630	0	−28	28	21	14	25	18	25	35			18
630	800	0	−35	35	26	18	30	20	30	42			20

Note: The tolerances of outer ring flange outer diameters are given in Table 1 - 43.

① Not applies to flange outer ring bearings.

② Only applies to groove ball bearings.

(4) Tolerance class 4 is given in Table 1-26 and Table 1-27.

Table 1-26 Inner ring

d/mm		Δ_{dmp} $\Delta_{\text{ds}}^{\textcircled{1}}$ /μm		$V_{\text{dsp}}/\mu\text{m}$		V_{dmp} /μm	K_{ia} /μm	S_{d} /μm	$S_{\text{ia}}^{\textcircled{2}}$ /μm	Δ_{Bs} /μm			V_{Bs} /μm
				diameter series						all	normal	modified ^③	
				9	0、1、 2、3、4								
over	incl.	upper	lower	max		max	max	max	max	upper	lower		max
—	0.6	0	−4	4	3	2	2.5	3	3	0	−40	−250	2.5
0.6	2.5	0	−4	4	3	2	2.5	3	3	0	−40	−250	2.5
2.5	10	0	−4	4	3	2	2.5	3	3	0	−40	−250	2.5
10	18	0	−4	4	3	2	2.5	3	3	0	−80	−250	2.5
18	30	0	−5	5	4	2.5	3	4	4	0	−120	−250	2.5
30	50	0	−6	6	5	3	4	4	4	0	−120	−250	3
50	80	0	−7	7	5	3.5	4	5	5	0	−150	−250	4
80	120	0	−8	8	6	4	5	5	5	0	−200	−380	4
120	180	0	−10	10	8	5	6	6	7	0	−250	−380	5
180	250	0	−12	12	9	6	8	7	8	0	−300	−500	6

- ① Only applies to diameter series 0, 1, 2, 3 and 4.
- ② Applies to groove type ball bearings only.
- ③ Applies to single bearing inner ring and outer ring of paired or stack mounting.

Table 1 – 27 Outer ring

D/mm		Δ_{Dmp} $\Delta_{\text{Ds}}^{①}$ $/\mu\text{m}$		$V_{\text{Dsp}}/\mu\text{m}$		V_{Dmp} $/\mu\text{m}$	K_{ea} $/\mu\text{m}$	$S_{\text{D}}^{②}$ $S_{\text{DI}}^{③}$ $/\mu\text{m}$	$S_{\text{ea}}^{②③}$ $/\mu\text{m}$	$S_{\text{eal}}^{③}$ $/\mu\text{m}$	Δ_{Cs} $\Delta_{\text{CIs}}^{③}$ $/\mu\text{m}$		V_{Cs} $V_{\text{CIs}}^{③}$ $/\mu\text{m}$
				diameter series									
				9	0、1、 2、3、4								
over	incl.	upper	lower	max		max	max	max	max	max	upper	lower	max
—	2.5	0	−4	4	3	2	3	4	5	7	identical to Δ_{Bs} of inner ring of the same bearing		2.5
2.5	6	0	−4	4	3	2	3	4	5	7			2.5
6	18	0	−4	4	3	2	3	4	5	7			2.5
18	30	0	−5	5	4	2.5	4	4	5	7			2.5
30	50	0	−6	6	5	3	5	4	5	7			2.5
50	80	0	−7	7	5	3.5	5	4	5	7			3
80	120	0	−8	8	6	4	6	5	6	8			4
120	150	0	−9	9	7	5	7	5	7	10			5
150	180	0	−10	10	8	5	8	5	8	11			5
180	250	0	−11	11	8	6	10	7	10	14			7
250	315	0	−13	13	10	7	11	8	10	14		7	
315	400	0	−15	15	11	8	13	10	13	18		8	

Note: The tolerances of outer ring flange outer diameters are given in Table 1 – 43.

① These deviations apply to diameter series 0, 1, 2, 3 and 4 only.

② Not applies to flange outer ring bearings.

③ Applies to groove type ball bearings only.

(5) Tolerance 2 is given in Table 1-28 and Table 1-29.

Table 1-28 Inner ring

d/mm		$\Delta_{\text{dmp}}/\mu\text{m}$ $\Delta_{\text{ds}}^{\text{①}}$		$V_{\text{dsp}}^{\text{①}}$ $/\mu\text{m}$	V_{dmp} $/\mu\text{m}$	K_{ia} $/\mu\text{m}$	S_{d} $/\mu\text{m}$	$S_{\text{ia}}^{\text{②}}$ $/\mu\text{m}$	$\Delta_{\text{Bs}}/\mu\text{m}$			V_{Bs} $/\mu\text{m}$
									all	normal	modified ^③	
over	incl.	upper	lower	max	max	max	max	max	upper	lower		max
—	0.6	0	-2.5	2.5	1.5	1.5	1.5	1.5	0	-40	-250	1.5
0.6	2.5	0	-2.5	2.5	1.5	1.5	1.5	1.5	0	-40	-250	1.5
2.5	10	0	-2.5	2.5	1.5	1.5	1.5	1.5	0	-40	-250	1.5
10	18	0	-2.5	2.5	1.5	1.5	1.5	1.5	0	-80	-250	1.5
18	30	0	-2.5	2.5	1.5	2.5	1.5	2.5	0	-120	-250	1.5
30	50	0	-2.5	2.5	1.5	2.5	1.5	2.5	0	-120	-250	1.5
50	80	0	-4	4	2	2.5	1.5	2.5	0	-150	-250	1.5
80	120	0	-5	5	2.5	2.5	2.5	2.5	0	-200	-380	2.5
120	150	0	-7	7	3.5	2.5	2.5	2.5	0	-250	-380	2.5
150	180	0	-7	7	3.5	5	4	5	0	-250	-380	4
180	250	0	-8	8	4	5	5	5	0	-300	-500	5

① Applies to diameter series 0, 1, 2, 3 and 4 only.

② Applies to groove type ball bearings only.

③ Applies to single bearing inner ring and outer ring of paired or stack mounting.

Table 1-29 Outer ring

D/mm		$\Delta_{\text{Dmp}}^{①}/\mu\text{m}$ $\Delta_{\text{Ds}}^{①}/\mu\text{m}$		$V_{\text{Dsp}}^{①}/\mu\text{m}$	$V_{\text{Dmp}}/\mu\text{m}$	$K_{\text{ea}}/\mu\text{m}$	$S_{\text{D}}^{②}/\mu\text{m}$ $S_{\text{Dl}}^{③}/\mu\text{m}$	$S_{\text{ea}}^{②③}/\mu\text{m}$	$S_{\text{eal}}^{③}/\mu\text{m}$	$\Delta_{\text{Cs}}/\mu\text{m}$ $\Delta_{\text{ClS}}^{③}/\mu\text{m}$		$V_{\text{Cs}}/\mu\text{m}$ $V_{\text{ClS}}^{③}/\mu\text{m}$
over	incl.	upper	lower	max	max	max	max	max	max	upper	lower	max
—	2.5	0	-2.5	2.5	1.5	1.5	1.5	1.5	3	Identical to Δ_{Bs} of inner ring of the same bearing		1.5
2.5	6	0	-2.5	2.5	1.5	1.5	1.5	1.5	3			1.5
6	18	0	-2.5	2.5	1.5	1.5	1.5	1.5	3			1.5
18	30	0	-4	4	2	2.5	1.5	2.5	4			1.5
30	50	0	-4	4	2	2.5	1.5	2.5	4			1.5
50	80	0	-4	4	2	4	1.5	4	6			1.5
80	120	0	-5	5	2.5	5	2.5	5	7			2.5
120	150	0	-5	5	2.5	5	2.5	5	7			2.5
150	180	0	-7	7	3.5	5	2.5	5	7			2.5
180	250	0	-8	8	4	7	4	7	10			4
250	315	0	-8	8	4	7	5	7	10			5
315	400	0	-10	10	5	8	7	8	11			7

Note: The tolerances of outer ring flange outer diameters are given in Table 1-43.

① These deviations apply to opened and capped bearings of diameter series 0, 1, 2, 3 and 4 only.

② Not applies to flange outer ring bearings.

③ Applies to groove type ball bearings only.

2. Tapered roller bearings

Bore diameter tolerances given in Table 1-30 ~ Table 1-42 apply to basically cylindrical bores.

(1) Tolerances of 0 class is given in Table 1-30 and Table 1-32.

Table 1-30 Inner ring

d/mm		$\Delta_{\text{dmp}}/\mu\text{m}$		$V_{\text{dsp}}/\mu\text{m}$	$V_{\text{dmp}}/\mu\text{m}$	$K_{\text{ia}}/\mu\text{m}$
over	incl.	upper	lower	max	max	max
—	10	0	-12	12	9	15
10	18	0	-12	12	9	15
18	30	0	-12	12	9	18
30	50	0	-12	12	9	20
50	80	0	-15	15	11	25
80	120	0	-20	20	15	30
120	180	0	-25	25	19	35
180	250	0	-30	30	23	50
250	315	0	-35	35	26	60
315	400	0	-40	40	30	70
400	500	0	-45	45	34	80
500	630	0	-60	60	40	90
630	800	0	-75	75	45	100
800	1000	0	-100	100	55	115
1000	1250	0	-125	125	65	130
1250	1600	0	-160	160	80	150
1600	2000	0	-200	200	100	170

Table 1 – 31 Outer ring

D/mm		$\Delta_{\text{Dmp}}/\mu\text{m}$		$V_{\text{Dsp}}/\mu\text{m}$	$V_{\text{Dmp}}/\mu\text{m}$	$K_{\text{ea}}/\mu\text{m}$
over	incl.	upper	lower	max	max	max
—	18	0	– 12	12	9	18
18	30	0	– 12	12	9	18
30	50	0	– 14	14	11	20
50	80	0	– 16	16	12	25
80	120	0	– 18	18	14	35
120	150	0	– 20	20	15	40
150	180	0	– 25	25	19	45
180	250	0	– 30	30	23	50
250	315	0	– 35	35	26	60
315	400	0	– 40	40	30	70
400	500	0	– 45	45	34	80
500	630	0	– 50	60	38	100
630	800	0	– 75	80	55	120
800	1000	0	– 100	100	75	140
1000	1250	0	– 125	130	90	160
1250	1600	0	– 160	170	100	180
1600	2000	0	– 200	210	110	200
2000	2500	0	– 250	265	120	220

Note: Tolerance for the outside diameter D_1 of an outer ring flange is given in Table 1 – 43.

Table 1 - 32 Width — Inner ring and outer ring, single row bearing and single row sub-units

d/mm		$\Delta_{Bs}/\mu\text{m}$		$\Delta_{Cs}/\mu\text{m}$		$\Delta_{Ts}/\mu\text{m}$		$\Delta_{T1s}/\mu\text{m}$		$\Delta_{T2s}/\mu\text{m}$	
over	incl.	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
—	10	0	-120	0	-120	+200	0	+100	0	+100	0
10	18	0	-120	0	-120	+200	0	+100	0	+100	0
18	30	0	-120	0	-120	+200	0	+100	0	+100	0
30	50	0	-120	0	-120	+200	0	+100	0	+100	0
50	80	0	-150	0	-150	+200	0	+100	0	+100	0
80	120	0	-200	0	-200	+200	-200	+100	-100	+100	-100
120	180	0	-250	0	-250	+350	-250	+150	-150	+200	-100
180	250	0	-300	0	-300	+350	-250	+150	-150	+200	-100
250	315	0	-350	0	-350	+350	-250	+150	-150	+200	-100
315	400	0	-400	0	-400	+400	-400	+200	-200	+200	-200
400	500	0	-450	0	-450	+450	-450	+225	-225	+225	-225
500	630	0	-500	0	-500	+500	-500	—	—	—	—
630	800	0	-750	0	-750	+600	-600	—	—	—	—
800	1000	0	-1000	0	-1000	+750	-750	—	—	—	—
1000	1250	0	-1250	0	-1250	+900	-900	—	—	—	—
1250	1600	0	-1600	0	-1600	+1050	-1050	—	—	—	—
1600	2000	0	-2000	0	-2000	+1200	-1200	—	—	—	—

(2) Tolerance class 6X The diameter and radial runout tolerances for inner and outer rings of this tolerance class are the same as those given in Table 1 – 30 and Table 1 – 31. Width tolerances are given in Table 1 – 33.

Table 1 – 33 Width — Inner ring and outer ring, single row bearing and single row sub-units

d/mm		$\Delta_{Bs}/\mu\text{m}$		$\Delta_{Cs}/\mu\text{m}$		$\Delta_{Ts}/\mu\text{m}$		$\Delta_{T1s}/\mu\text{m}$		$\Delta_{T2s}/\mu\text{m}$	
over	incl.	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
—	10	0	– 50	0	– 100	+ 100	0	+ 50	0	+ 50	0
10	18	0	– 50	0	– 100	+ 100	0	+ 50	0	+ 50	0
18	30	0	– 50	0	– 100	+ 100	0	+ 50	0	+ 50	0
30	50	0	– 50	0	– 100	+ 100	0	+ 50	0	+ 50	0
50	80	0	– 50	0	– 100	+ 100	0	+ 50	0	+ 50	0
80	120	0	– 50	0	– 100	+ 100	0	+ 50	0	+ 50	0
120	180	0	– 50	0	– 100	+ 150	0	+ 50	0	+ 100	0
180	250	0	– 50	0	– 100	+ 150	0	+ 50	0	+ 100	0
250	315	0	– 50	0	– 100	+ 200	0	+ 100	0	+ 100	0
315	400	0	– 50	0	– 100	+ 200	0	+ 100	0	+ 100	0
400	500	0	– 50	0	– 100	+ 200	0	+ 100	0	+ 100	0

(3) Tolerance class 5 is given in Table 1-34 ~ Table 1-36.

Table 1-34 Inner ring

d/mm		$\Delta_{\text{dmp}}/\mu\text{m}$		$V_{\text{dsp}}/\mu\text{m}$	$V_{\text{dmp}}/\mu\text{m}$	$K_{\text{ia}}/\mu\text{m}$	$S_{\text{d}}/\mu\text{m}$
over	incl.	upper	lower	max	max	max	max
—	10	0	-7	5	5	5	7
10	18	0	-7	5	5	5	7
18	30	0	-8	6	5	5	8
30	50	0	-10	8	5	6	8
50	80	0	-12	9	6	7	8
80	120	0	-15	11	8	8	9
120	180	0	-18	14	9	11	10
180	250	0	-22	17	11	13	11
250	315	0	-25	19	13	13	13
315	400	0	-30	23	15	15	15
400	500	0	-35	28	17	20	17
500	630	0	-40	35	20	25	20
630	800	0	-50	45	25	30	25
800	1000	0	-60	60	30	37	30
1000	1250	0	-75	75	37	45	40
1250	1600	0	-90	90	45	55	50

Table 1 – 35 Outer ring

D/mm		$\Delta_{\text{Dmp}}/\mu\text{m}$		$V_{\text{Dsp}}/\mu\text{m}$	$V_{\text{Dmp}}/\mu\text{m}$	$K_{\text{ea}}/\mu\text{m}$	$S_{\text{D}}^{\text{①}}、S_{\text{D1}}/\mu\text{m}$
over	incl.	upper	lower	max	max	max	max
—	18	0	− 8	6	5	6	8
18	30	0	− 8	6	5	6	8
30	50	0	− 9	7	5	7	8
50	80	0	− 11	8	6	8	8
80	120	0	− 13	10	7	10	9
120	150	0	− 15	11	8	11	10
150	180	0	− 18	14	9	13	10
180	250	0	− 20	15	10	15	11
250	315	0	− 25	19	13	18	13
315	400	0	− 28	22	14	20	13
400	500	0	− 33	26	17	24	17
500	630	0	− 38	30	20	30	20
630	800	0	− 45	38	25	36	25
800	1000	0	− 60	50	30	43	30
1000	1250	0	− 80	65	38	52	38
1250	1600	0	− 100	90	50	62	50
1600	2000	0	− 125	120	65	73	65

Note: The tolerance for the outside diameter D_1 of an outer ring flange is given in Table 1 – 43.

① Not applies to flange outer ring bearings.

Table 1-36 Width — Inner ring and outer ring, and single row bearing and single row sub-units

d/mm		$\Delta_{Bs}/\mu\text{m}$		$\Delta_{Cs}/\mu\text{m}$		$\Delta_{Ts}/\mu\text{m}$		$\Delta_{T1s}/\mu\text{m}$		$\Delta_{T2s}/\mu\text{m}$	
over	incl.	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
—	10	0	-200	0	-200	+200	-200	+100	-100	+100	-100
10	18	0	-200	0	-200	+200	-200	+100	-100	+100	-100
18	30	0	-200	0	-200	+200	-200	+100	-100	+100	-100
30	50	0	-240	0	-240	+200	-200	+100	-100	+100	-100
50	80	0	-300	0	-300	+200	-200	+100	-100	+100	-100
80	120	0	-400	0	-400	+200	-200	+100	-100	+100	-100
120	180	0	-500	0	-500	+350	-250	+150	-150	+200	-100
180	250	0	-600	0	-600	+350	-250	+150	-150	+200	-100
250	315	0	-700	0	-700	+350	-250	+150	-150	+200	-100
315	400	0	-800	0	-800	+400	-400	+200	-200	+200	-200
400	500	0	-900	0	-900	+450	-450	+225	-225	+225	-225
500	630	0	-1100	0	-1100	+500	-500	—	—	—	—
630	800	0	-1600	0	-1600	+600	-600	—	—	—	—
800	1000	0	-2000	0	-2000	+750	-750	—	—	—	—
1000	1250	0	-2000	0	-2000	+750	-750	—	—	—	—
1250	1600	0	-2000	0	-2000	+900	-900	—	—	—	—

(4) Tolerance class 4 is given in Table 1 – 37 ~ Table 1 – 39.

Table 1 – 37 Inner ring

d/mm		$\begin{matrix} \Delta_{\text{dmp}}/\mu\text{m} \\ \Delta_{\text{ds}} \end{matrix}$		$V_{\text{dsp}}/\mu\text{m}$	$V_{\text{dmp}}/\mu\text{m}$	$K_{\text{ia}}/\mu\text{m}$	$S_{\text{d}}/\mu\text{m}$	$S_{\text{ia}}/\mu\text{m}$
over	incl.	upper	lower	max	max	max	max	max
—	10	0	– 5	4	4	3	3	3
10	18	0	– 5	4	4	3	3	3
18	30	0	– 6	5	4	3	4	4
30	50	0	– 8	6	5	4	4	4
50	80	0	– 9	7	5	4	5	4
80	120	0	– 10	8	5	5	5	5
120	180	0	– 13	10	7	6	6	7
180	250	0	– 15	11	8	8	7	8
250	315	0	– 18	12	9	9	8	9

Table 1-38 Outer ring

d/mm		$\begin{matrix} \Delta_{\text{Dmp}}/\mu\text{m} \\ \Delta_{\text{Ds}} \end{matrix}$		$V_{\text{Dsp}}/\mu\text{m}$	$V_{\text{Dmp}}/\mu\text{m}$	$K_{\text{ea}}/\mu\text{m}$	$\begin{matrix} S_{\text{D}}^{\text{①}} \\ S_{\text{D1}} \end{matrix}/\mu\text{m}$	$S_{\text{ea}}^{\text{①}}/\mu\text{m}$	$S_{\text{eal}}/\mu\text{m}$
over	incl.	upper	lower	max	max	max	max	max	max
—	18	0	-6	5	4	4	4	5	7
18	30	0	-6	5	4	4	4	5	7
30	50	0	-7	5	5	5	4	5	7
50	80	0	-9	7	5	5	4	5	7
80	120	0	-10	8	5	6	5	6	8
120	150	0	-11	8	6	7	5	7	10
150	180	0	-13	10	7	8	5	8	11
180	250	0	-15	11	8	10	7	10	14
250	315	0	-18	14	9	11	8	10	14
315	400	0	-20	15	10	13	10	13	18

Note: The tolerance for the outside diameter D_1 of an outer ring flange is given in Table 1-43.

① Not applies to flange outer ring bearings.

Table 1 – 39 Width — Inner ring and outer ring, and single row bearing and single row sub-units

d/mm		$\Delta_{Bs}/\mu\text{m}$		$\Delta_{Cs}/\mu\text{m}$		$\Delta_{Ts}/\mu\text{m}$		$\Delta_{T1s}/\mu\text{m}$		$\Delta_{T2s}/\mu\text{m}$	
over	incl.	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
—	10	0	– 200	0	– 200	+ 200	– 200	+ 100	– 100	+ 100	– 100
10	18	0	– 200	0	– 200	+ 200	– 200	+ 100	– 100	+ 100	– 100
18	30	0	– 200	0	– 200	+ 200	– 200	+ 100	– 100	+ 100	– 100
30	50	0	– 240	0	– 240	+ 200	– 200	+ 100	– 100	+ 100	– 100
50	80	0	– 300	0	– 300	+ 200	– 200	+ 100	– 100	+ 100	– 100
80	120	0	– 400	0	– 400	+ 200	– 200	+ 100	– 100	+ 100	– 100
120	180	0	– 500	0	– 500	+ 350	– 250	+ 150	– 150	+ 200	– 100
180	250	0	– 600	0	– 600	+ 350	– 250	+ 150	– 150	+ 200	– 100
250	315	0	– 700	0	– 700	+ 350	– 250	+ 150	– 150	+ 200	– 100

(5) Tolerance class 2 is given in Table 1 – 40 ~ Table 1 – 42.

Table 1 – 40 Inner ring

d/mm		$\begin{matrix} \Delta_{\text{dmp}}/\mu\text{m} \\ \Delta_{\text{ds}} \end{matrix}$		$V_{\text{dsp}}/\mu\text{m}$	$V_{\text{dmp}}/\mu\text{m}$	$K_{\text{ia}}/\mu\text{m}$	$S_{\text{d}}/\mu\text{m}$	$S_{\text{ia}}/\mu\text{m}$
over	incl.	upper	lower	max	max	max	max	max
—	10	0	–4	2.5	1.5	2	1.5	2
10	18	0	–4	2.5	1.5	2	1.5	2
18	30	0	–4	2.5	1.5	2.5	1.5	2.5
30	50	0	–5	3	2	2.5	2	2.5
50	80	0	–5	4	2	3	2	3
80	120	0	–6	5	2.5	3	2.5	3
120	180	0	–7	7	3.5	4	3.5	4
180	250	0	–8	7	4	5	5	5
250	315	0	–8	8	5	6	5.5	6

Table 1 – 41 Outer ring

D/mm		$\Delta_{\text{Dmp}}/\mu\text{m}$ Δ_{Ds}		$V_{\text{Dsp}}/\mu\text{m}$	$V_{\text{Dmp}}/\mu\text{m}$	$K_{\text{ea}}/\mu\text{m}$	$S_{\text{D}}^{\text{①}}$ $S_{\text{D1}}/\mu\text{m}$	$S_{\text{ea}}^{\text{①}}/\mu\text{m}$	$S_{\text{eal}}/\mu\text{m}$
over	incl.	upper	lower	max	max	max	max	max	max
—	18	0	– 5	4	2. 5	2. 5	1. 5	2. 5	4
18	30	0	– 5	4	2. 5	2. 5	1. 5	2. 5	4
30	50	0	– 5	4	2. 5	2. 5	2	2. 5	4
50	80	0	– 6	4	2. 5	4	2. 5	4	6
80	120	0	– 6	5	3	5	3	5	7
120	150	0	– 7	5	3. 5	5	3. 5	5	7
150	180	0	– 7	7	4	5	4	5	7
180	250	0	– 8	8	5	7	5	7	10
250	315	0	– 9	8	5	7	6	7	10
315	400	0	– 10	10	6	8	7	8	11

Note: The tolerance for the outside diameter D_1 of an outer ring flange is given in Table 1 – 43.

① Not applies to flange outer ring bearings.

Table 1 - 42 Width-Inner ring and outer ring, and single row bearing and single row sub-units

d/mm		$\Delta_{Bs}/\mu\text{m}$		$\Delta_{Cs}/\mu\text{m}$		$\Delta_{Ts}/\mu\text{m}$		$\Delta_{T1s}/\mu\text{m}$		$\Delta_{T2s}/\mu\text{m}$	
over	incl.	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
—	10	0	-200	0	-200	+200	-200	+100	-100	+100	-100
10	18	0	-200	0	-200	+200	-200	+100	-100	+100	-100
18	30	0	-200	0	-200	+200	-200	+100	-100	+100	-100
30	50	0	-240	0	-240	+200	-200	+100	-100	+100	-100
50	80	0	-300	0	-300	+200	-200	+100	-100	+100	-100
80	120	0	-400	0	-400	+200	-200	+100	-100	+100	-100
120	180	0	-500	0	-500	+200	-250	+100	-100	+100	-150
180	250	0	-600	0	-600	+200	-300	+100	-150	+100	-150
250	315	0	-700	0	-700	+200	-300	+100	-150	+100	-150

3. Outer ring flange of radial rolling bearings

The tolerances of outside diameter of outer ring flange given in Table 1 – 43 apply to radial rolling bearings and tapered roller bearings.

Table 1 – 43 Tolerances of outside flange diameter

D_1 /mm		$\Delta_{Dis}/\mu\text{m}$			
		locating flange		non-locating flange	
over	incl.	upper	lower	upper	lower
—	6	0	– 36	+ 220	– 36
6	10	0	– 36	+ 220	– 36
10	18	0	– 43	+ 270	– 43
18	30	0	– 52	+ 330	– 52
30	50	0	– 62	+ 390	– 62
50	80	0	– 74	+ 460	– 74
80	120	0	– 87	+ 540	– 87
120	180	0	– 100	+ 630	– 100
180	250	0	– 115	+ 720	– 115
250	315	0	– 130	+ 810	– 130
315	400	0	– 140	+ 890	– 140
400	500	0	– 155	+ 970	– 155
500	630	0	– 175	+ 1100	– 175
630	800	0	– 200	+ 1250	– 200
800	1000	0	– 230	+ 1400	– 230
1000	1250	0	– 260	+ 1650	– 260
1250	1600	0	– 310	+ 1950	– 310
1600	2000	0	– 370	+ 2300	– 370
2000	2500	0	– 440	+ 2800	– 440

4. Basically tapered bore, taper 1:12 and 1:30

Figure 1 – 34 is the schematic view of a theoretical tapered bore. Figure 1 – 35 is the schematic view of a tapered bore with actual mean diameters and their deviations.

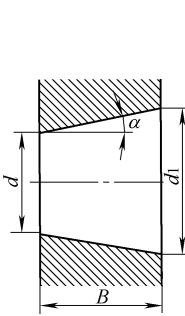


Figure 1 – 34 Theoretical tapered bore

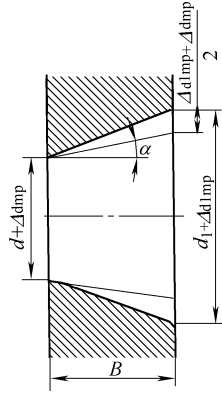


Figure 1 – 35 Tapered bore with actual diameters and deviations

(1) Taper 1:12 The taper angle (half taper angle) is:

$$\alpha = 2^{\circ}23'9.4'' = 2.38594^{\circ} = 0.041643\text{rad}$$

Basic diameter at the large end of the theoretical bore is:

$$d_1 = d + \frac{1}{12}B$$

(2) Taper 1:30 The taper angle (half taper angle) is:

$$\alpha = 0^{\circ}57'17.4'' = 0.95484^{\circ} = 0.016665\text{rad}$$

Basic diameter at the large end of the theoretical bore is:

$$d_1 = d + \frac{1}{30}B$$

The tolerance of a tapered bore includes:

1) The tolerance of tapered bore mean diameter is expressed by the limit Δ_{dmp} of the deviation of the theoretical small end mean diameter.

2) The tolerance of taper is expressed by the limit of difference ($\Delta_{\text{d1mp}} - \Delta_{\text{dmp}}$) of the deviations of the mean diameters of the two ends of the bore.

3) The tolerance of the variation of diameter is expressed by maximum value V_{dsp} of the bore in any radial plane.

Tolerance class 0 of the tapered bore is given in Table 1 – 44 and Table 1 – 45.

Table 1 – 44 Tapered bore (taper 1:12)

d/mm		$\Delta_{\text{dmp}}/\mu\text{m}$		$\Delta_{\text{dlmp}} - \Delta_{\text{dmp}}/\mu\text{m}$		$V_{\text{dsp}}^{①,②}/\mu\text{m}$
over	incl.	upper	lower	upper	lower	max
—	10	+ 22	0	+ 15	0	9
10	18	+ 27	0	+ 18	0	11
18	30	+ 33	0	+ 21	0	13
30	50	+ 39	0	+ 25	0	16
50	80	+ 46	0	+ 30	0	19
80	120	+ 54	0	+ 35	0	22
120	180	+ 63	0	+ 40	0	40
180	250	+ 72	0	+ 46	0	46
250	315	+ 81	0	+ 52	0	52
315	400	+ 89	0	+ 57	0	57
400	500	+ 97	0	+ 63	0	63
500	630	+ 110	0	+ 70	0	70
630	800	+ 125	0	+ 80	0	—
800	1000	+ 140	0	+ 90	0	—
1000	1250	+ 165	0	+ 105	0	—
1250	1600	+ 195	0	+ 125	0	—

① Applies for any single radial plane of bores.

② Not apply for diameter series 7 and 8.

Table 1 – 45 Tapered bore (taper 1:30)

d/mm		$\Delta_{\text{dmp}}/\mu\text{m}$		$\Delta_{\text{dlmp}} - \Delta_{\text{dmp}}/\mu\text{m}$		$V_{\text{dsp}}^{①,②}/\mu\text{m}$
over	incl.	upper	lower	upper	lower	max
—	50	+ 15	0	+ 30	0	19
50	80	+ 15	0	+ 30	0	19
80	120	+ 20	0	+ 35	0	22
120	180	+ 25	0	+ 40	0	40
180	250	+ 30	0	+ 46	0	46
250	315	+ 35	0	+ 52	0	52
315	400	+ 40	0	+ 57	0	57
400	500	+ 45	0	+ 63	0	63
500	630	+ 50	0	+ 70	0	70

① Applies for any single radial plane of bores.

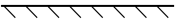
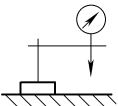
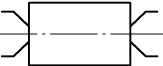
② Not apply for diameter series 7 and 8.

1.4.4 Inspection Method of Rolling Bearing Tolerance and Rotating Accuracy

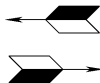
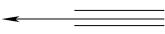
1. General condition of measurement

The schematic symbols used in this section are shown in Table 1 – 46.

Table 1 – 46 Schematic symbols

symbol	illustration
	level plate (measuring plane)
 (main view)  (top view or bottom view)	fixed support point
	fixed measuring support point
 (main view)  (top view or bottom view)	indicator or recorder
	measuring support with indicator or recorder According to the type of measuring equipment, the measuring support symbols can be drawn as different types.
	locating mandrel
	intermittent reciprocating line motion
	rotating motion rely on a fixed support point

(continued)

symbol	illustration
	rotate around a center
	load、load direction
	alternating load at opposite directions
 (main view)  (top view or bottom view)	moving support point of moving indicator that is perpendicular to the measured surface
	moving support point of moving indicator that is parallel to the measured surface

1) The measurement of all dimensions and runouts can be performed on different types of measuring equipment that can have different accuracy. All of the measuring errors must be not more than 10% of the actual tolerance band in principle.

2) The dimension can be determined by comparing the actual measuring part with the corresponding gauge or standard part. The corresponding gauge and standard part must be calibrated and transferred according to stipulations. This measurement must use a calibrated indicator instrument with proper sensitivity.

3) Mandrel. While measuring the runout by using a mandrel, in order to correct the mandrel error in afterward bearing measurement, it must be determined firstly that what the running accuracy of the mandrel is. A

precision mandrel with the taper about 1:5000 can be used in this case.

While measuring the overall inner diameter of the rollers by using a mandrel, a precision mandrel with the taper of 1:2000 can be used.

4) Temperature. Before measuring, the measured parts, the measuring instruments, and the standard parts should have the same temperature as the measurement room. The recommended temperature is $+20^{\circ}\text{C}$. During measuring, it should be prevented as far as possible that the heat would transfer to the parts or assembled bearings.

5) The measuring force and the radius of the measuring tip. In order to avoid the excessive deformation of the thin wall of bearing rings, the measuring force should be reduced as small as possible. If there is an apparent deformation, it is better to introduce a modification factor of the load deformation that could modify the measuring value as the value under a free, no-load condition. The largest measuring force and the smallest measuring tip radius are shown in Table 1-47.

Table 1-47 The largest measuring force and the smallest measuring tip radius

bearing position	nominal dimension range /mm		measuring force ^① /N	measuring tip radius ^② /mm
	over	incl.	max	min
inner ring d	—	10	2	0.8
	10	30	2	2.5
	30	—	2	2.5
outer ring D	—	30	2	2.5
	30	—	2	2.5

① The largest measuring force should be the one under which the measurement result could be repeated.

② Along with proper decrease of the measuring force, a smaller tip radius could be used.

6) Central axial measuring load. In order to keep the normal relative positions of each bearing part, the central axial measuring load should be used for some stipulated measuring methods.

Table 1 – 48 The central axial measuring load for the radial ball bearing and the angular contact ball bearing of the contact angle $\leq 30^{\circ}\text{C}$

outer diameter D/mm		central axial load on bearing/ N
over	incl.	min
—	30	5
30	50	10
50	80	20
80	120	35
120	180	70
180	—	140

Table 1 – 49 The central axial measuring load for the tapered roller bearing, the angular contact ball bearing of the contact angle $> 30^{\circ}\text{C}$ and the thrust bearing

outer diameter D/mm		central axial load on bearing/ N
over	incl.	min
—	30	40
30	50	80
50	80	120
80	120	150
120	—	150

7) Measuring region limit. The measuring of the deviation limit of the inner ring diameter or outer ring diameter should be performed in the radial plane that must be apart from the ring end face or flange end face for the distance that is larger than a . The values of a are shown in Table 1 – 50. The maximum solid dimension is only suitable for outside of the measuring region limit.

Table 1 – 50 Measuring region limit (unit: mm)

r_{min}		a
over	incl.	
—	0.6	$r_{\text{max}} + 0.5$
0.6	—	$1.2 * r_{\text{max}}$

8) Preparation before measuring. The grease or the anti-rusting agent adhered on bearings that could influence the measuring result should be cleared away. Before measuring, bearings should be lubricated by oil with low viscosity. Some structures of the pre-lubricated bearings, the sealed bearings, and the shielded bearings may have some influence on the measuring accuracy. In order to eliminate the difference, the measurement should be performed on open bearings after dismantling the seal /shield and clearing away the lubricant.

Note: After finishing measuring, the bearings should be anti-rusted immediately.

9) The standard base surface of measurement. The standard base surfaces are assigned by the manufacturer, usually, which can be the measuring datum base.

Note: The standard measuring base surface of the rings usually is a non-marking surface. When it is not certain for the symmetrical rings which one is the standard measuring base surface, the tolerance can be suitable for any end face.

The standard base surface of the shaft washer and the housing washer of thrust bearings, usually, is the end face that can be subjected to axial load, which usually is the back face of the raceway.

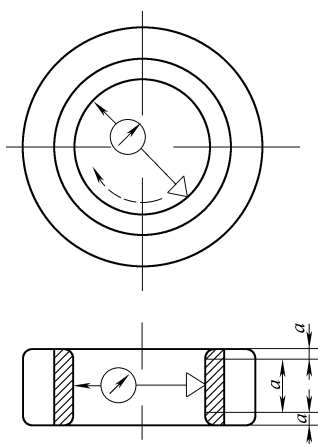
The standard base surface of single row angular contact ball bearings and tapered roller bearings is the end face subjected to axial load.

The standard base surface of flange outer ring bearings is the flange end face subjected to axial load.

2. The principle of the measurement of the single bore diameter

(1) The measurement of the single bore diameter The measurement method of the single bore diameter is shown in Table 1 – 51.

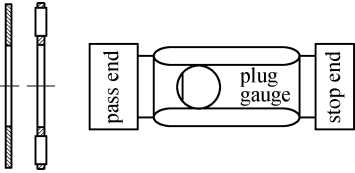
Table 1 – 51 The measurement method of the single bore diameter

method	illustration
 a — measuring region Calibrating the instrument to zero by a proper gauge or a standard ring; Within the measuring region, in a single radial plane and in several angle directions, measuring and recording the maximum and minimum single bore diameters d_{spmax} and d_{spmin}. Within several radial planes, repeating measurement and recording the results, in order to determine the maximum and minimum single bore diameter d_{smax} and d_{smin}.	This method applies to all kinds of rolling bearing ring, shaft washer and central washer. The single bore diameter d_{sp} or d_s can be read directly from the indicator. This method applies also for measuring the inner diameter of outer ring of the separable cylindrical roller bearing or needle roller bearing. But it must be avoided to put the measuring point on the guiding chamfer of the raceway. In order to avoid the influence of the gravitational force, the axis of the bearing ring or washer should be on the vertical position. According to the measuring values of d_{spmax} and d_{spmin}, the following values can be obtained: d_{mp} — the mean bore diameter in a single plane Δ_{dmp} — the deviation of the mean bore diameter in a single plane V_{dsp} — the variation of the single plane bore diameter V_{dmp} — the mean bore diameter variation According to the measuring values of d_s, d_{smax} and d_{smin}, the following values can be obtained: d_m — mean bore diameter Δ_{dm} — mean bore diameter deviation Δ_{ds} — single bore diameter deviation V_{ds} — bore diameter variation

(2) Function inspection of the smallest single bore diameter of the thrust needle roller and cage assembly and the thrust washer

The function inspection method of the smallest single bore diameter of the thrust needle roller and cage assembly and the thrust washer is shown in Table 1 – 52.

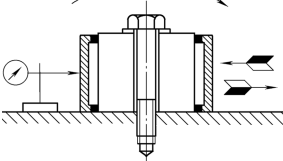
Table 1 – 52 The function inspection method of the smallest single bore diameter of the thrust needle roller and cage assembly and the thrust washer

method	illustration
 Under free condition, using the pass end and stop end of the plug gauge to measure the bore diameter of the thrust needle roller and cage assembly and the thrust washer. The pass end dimension of the plug gauge is the smallest bore diameter of the thrust needle roller and cage assembly and the thrust washer stipulating in the GB/T 4605—2003 as $d_{c\min}$ or $d_{s\min}$. The plug gauge stop end dimensions is the largest bore diameter of the thrust needle roller and cage assembly and thrust washer, stipulating in the GB/T 4605—2003.	This method can apply to the thrust needle roller and cage assembly and thrust washer stipulating in the GB/T 4605—2003. This method can also apply to the measuring of the smallest bore diameter $D_{l\min}$ of the housing washer stipulating in the GB/T 273.2—1998. By its own gravity, the assembly or the washer should be dropped freely from the pass end of the plug gauge. The stop end of the plug gauge should be not inserted in the bore of the assembly or the washer. If the stop end of the plug gauge could be inserted in the bore of the assembly or the washer forcibly, the assembly or the washer should be not dropped by its own gravity. The plug gauge applies to the inspection of the dimension limit only, not applies to measuring the bore diameter directly. Note: Because the tolerance of the thrust needle roller and cage assembly is different from that of the relevant washer respectively, different plug gauges should be used.

(3) Measurement of the single rolling element complement bore diameter

The measurement method of the single roller complement bore diameter is shown in Table 1 – 53.

Table 1 – 53 The measurement method of the single bore diameter of the rolling element complement

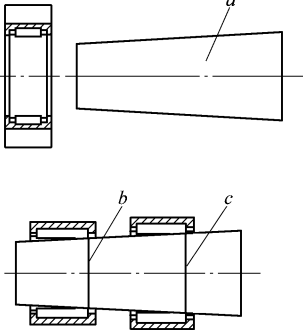
method	illustration
	This method can apply to the cylindrical roller bearings, needle roller bearings and drawn-cup needle roller bearings without inner ring. The single roller complement diameter F_{ws} equals to the measuring value plus the diameter of the standard measuring gauge.

(continued)

method	illustration																																															
Fix the standard measuring gauge onto the even platform. Measuring on the free condition for machined ring bearing. For the drawn cup needle roller bearing, first press the bearing into a hardening steel round gauge. The inner diameter of the hardening steel round gauge is stipulated in the JB/T 8878 — 2001. The smallest section dimensions of the round gauge are shown in the illustration. The bearing is slipped onto the standard measuring gauge, and the indicator should indicate the outside surface of middle part on the outer ring width along with the radial direction. On the same radial direction of the indicator, for outer ring apply the enough reciprocating radial load and measure the radial movement of the outer ring. The applying radial load value are shown on the table. Record the reading on the radial limit position of the outer ring. Rotate the bearing, then repeat measuring on several positions, so as to decide the largest and the smallest F_{wsmax} and F_{wsmin}.	According to the F_{wsmax} and F_{wsmin}, we can get: F_{wm} is the mean diameter of the roller complement; ΔF_{wm} is the deviation of the roller complement mean diameter. The smallest cross section dimension of the round gauge for the drawn cup needle roller bearing <table><tr><th colspan="2">nominal diameter of the round gauge inner ring/mm</th><th>cross section dimension of the round gauge /mm</th></tr><tr><th>over</th><th>incl.</th><th>min^①</th></tr><tr><td>6</td><td>10</td><td>10</td></tr><tr><td>10</td><td>18</td><td>12</td></tr><tr><td>18</td><td>30</td><td>15</td></tr><tr><td>30</td><td>50</td><td>18</td></tr><tr><td>50</td><td>80</td><td>20</td></tr><tr><td>80</td><td>120</td><td>25</td></tr><tr><td>120</td><td>150</td><td>30</td></tr></table> ① For precise measuring, we could use larger cross section dimension of the round gauge. The radial measuring load <table><tr><th colspan="2">F_w/mm</th><th>measuring load/N</th></tr><tr><th>over</th><th>incl.</th><th>min</th></tr><tr><td>—</td><td>30</td><td>50</td></tr><tr><td>30</td><td>50</td><td>60</td></tr><tr><td>50</td><td>80</td><td>70</td></tr><tr><td>80</td><td>—</td><td>80</td></tr></table>			nominal diameter of the round gauge inner ring/mm		cross section dimension of the round gauge /mm	over	incl.	min ^①	6	10	10	10	18	12	18	30	15	30	50	18	50	80	20	80	120	25	120	150	30	F_w /mm		measuring load/N	over	incl.	min	—	30	50	30	50	60	50	80	70	80	—	80
nominal diameter of the round gauge inner ring/mm		cross section dimension of the round gauge /mm																																														
over	incl.	min ^①																																														
6	10	10																																														
10	18	12																																														
18	30	15																																														
30	50	18																																														
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80	120	25																																														
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F_w /mm		measuring load/N																																														
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30	50	60																																														
50	80	70																																														
80	—	80																																														

(4) The measurement method of the smallest single bore diameter of the rolling element complement The measurement method of the smallest single bore diameter of the rolling element complement is shown in Table 1 - 54.

Table 1 - 54 The measurement method of the smallest single bore diameter of the rolling element complement

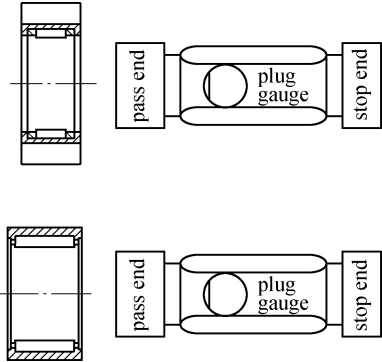
method	illustration																											
 a — taper mandrel b — the smallest calibrated diameter c — the largest calibrated diameter The measurement of the rolling element complement bore diameter is carried on by a round, calibrated taper mandrel. This taper mandrel has included all ranges of the bore diameter dimensions, and its taper is about 1:2000. For machined ring bearing, measuring is under its free condition. For the drawn cup needle roller bearing, first we press the bearing into a hardening steel round gauge, the inner diameter of which which is stipulated in the JB/T 8878-2001. The smallest cross section dimension of the round gauge is shown in the illustration.	This method can apply to all of the cylindrical roller bearings without inner ring, the needle roller bearings without inner ring and the drawn cup needle roller bearings that $F_w \leq 150\text{mm}$. This method can apply to measuring the smallest single bore diameter F_{wsmin} of the rolling element complement. But it can not measure the single bore diameter of the rolling element complement directly F_{ws}. This method also can apply to inspection. At the diameter place of the tolerance range limit of the bearing bore diameter, we mark the mandrel. If the mandrel diameter at the position contacting with the rolling element complement is over the smallest diameter marking line, but not over the largest diameter marking line, this bore diameter tolerance limit of the rolling element complement meets the requirement. The smallest cross section dimension of the round gauge for the drawn cup needle roller bearing <table><tr><th colspan="2">nominal diameter of the round gauge/mm</th><th>the cross section dimension of the round gauge/mm</th></tr><tr><th>over</th><th>incl.</th><th>min^①</th></tr><tr><td>6</td><td>10</td><td>10</td></tr><tr><td>10</td><td>18</td><td>12</td></tr><tr><td>18</td><td>30</td><td>15</td></tr><tr><td>30</td><td>50</td><td>18</td></tr><tr><td>50</td><td>80</td><td>20</td></tr><tr><td>80</td><td>120</td><td>25</td></tr><tr><td>120</td><td>150</td><td>30</td></tr></table> ① For precise measuring, we could use larger cross section dimension of the round gauge.	nominal diameter of the round gauge/mm		the cross section dimension of the round gauge/mm	over	incl.	min ^①	6	10	10	10	18	12	18	30	15	30	50	18	50	80	20	80	120	25	120	150	30
nominal diameter of the round gauge/mm		the cross section dimension of the round gauge/mm																										
over	incl.	min ^①																										
6	10	10																										
10	18	12																										
18	30	15																										
30	50	18																										
50	80	20																										
80	120	25																										
120	150	30																										

(continued)

method	illustration		
We insert the taper mandrel into the bearing bore and make a little vibrating for eliminating the radial clearance and adjusting the roller position but not letting the bearing expanding. The inserting mandrel axial load can be found in the illustration. We pull out the mandrel, and when the complement is located at the largest mandrel diameter, measure its diameter. Note: Before measuring, We could smear over a thin layer of anti-rusting agent on the bearing in order to show the precise stopping position of the rolling element on the mandrel.	The axial inserting load when measuring with taper mandrel		
	F_w/mm		axial load ^①
	over	incl.	/N
	8	15	10
	15	30	15
	30	80	30
	80	150	50
	① If there is no influence on measuring, also we can apply larger load.		

(5) Function inspection of the smallest single bore diameter of the rolling element complement

Table 1 – 55 Function inspection of the smallest single bore diameter of the rolling element complement

method	illustration
	This method can apply to all of the cylindrical roller bearings without inner ring, the needle roller bearings without inner ring and the drawn cup needle roller bearings that $F_w \leq 150\text{mm}$. By its own gravity of the bearing (for the drawn cup needle roller bearing that has been inserted to the round gauge, by the tatol weight of the round gauge and the bearing), it should drop freely from the pass end of the plug gauge, but it should not drop freely from the stop end of the plug gauge.

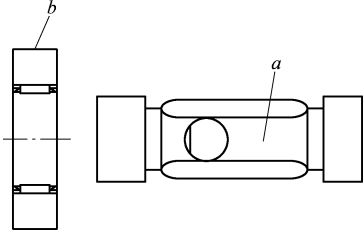
(continued)

method	illustration																													
The rolling element complement bore diameter F_w is inspected by the pass end and the stop end of the plug gauge. The machined ring bearing is measured under its free condition. For the drawn cup needle roller bearing, first we press the bearing into a hardening steel round gauge, the inner diameter of which is stipulated in the JB/T 8878—2001. The smallest cross section dimension of the round gauge is shown in the illustration. Then, the rolling element bore diameter is inspected by the pass end and the stop end of the plug gauge. The pass end dimension of the plug gauge is the smallest bore diameter of the rolling element complement. The stop end dimension of the plug gauge is 0.002mm larger than the largest bore diameter of the rolling element complement.	The plug gauge applies to inspecting the dimension limit only, but not applies to measuring the rolling element complement bore diameter F_{ws} directly. This inspecting method can confirm whether the range of F_{wsmin} is in the range of the tolerance limit. The smallest cross section dimension of the round gauge for the drawn cup needle roller bearing <table><tr><th colspan="2">nominal diameter of the round gauge inner ring/mm</th><th>cross section dimension of the round gauge/mm</th></tr><tr><th>over</th><th>incl.</th><th>min^①</th></tr><tr><td>6</td><td>10</td><td>10</td></tr><tr><td>10</td><td>18</td><td>12</td></tr><tr><td>18</td><td>30</td><td>12</td></tr><tr><td>30</td><td>50</td><td>18</td></tr><tr><td>50</td><td>80</td><td>20</td></tr><tr><td>80</td><td>120</td><td>25</td></tr><tr><td>120</td><td>150</td><td>30</td></tr></table>			nominal diameter of the round gauge inner ring/mm		cross section dimension of the round gauge/mm	over	incl.	min ^①	6	10	10	10	18	12	18	30	12	30	50	18	50	80	20	80	120	25	120	150	30
nominal diameter of the round gauge inner ring/mm		cross section dimension of the round gauge/mm																												
over	incl.	min ^①																												
6	10	10																												
10	18	12																												
18	30	12																												
30	50	18																												
50	80	20																												
80	120	25																												
120	150	30																												
① For precise measuring, we could use larger round gauge cross section dimension.																														

(6) Function inspection of the smallest single bore diameter of the rolling element complement (the radial needle roller and cage assembly)

The smallest single bore diameter function inspection (the radial needle roller and assembly) method of the rolling element complement is shown in Table 1 - 56.

Table 1 – 56 The smallest single bore diameter function inspection (the radial needle roller and cage assembly) method of the rolling element complement

method	illustration
 <i>a</i> — plug gauge <i>b</i> — round gauge Put the radial needle roller and cage assembly into a round gauge, the outside raceway dimension of which is stimulated in the JB/ T 7918—1997. The round gauge dimension equals to the nominal outside diameter E_w of the rolling element complement plus the lower deviation of the tolerance class G6 (see GB/T1800.2—1998). Insert the plug gauge, whose dimension equals to the nominal rolling element complement bore diameter F_w stipulated in the JB/T7918—1997. While the round gauge and the plug gauge making relative rotating each other, the radial needle roller and cage assembly should rotate freely.	This method can apply to radial needle roller and cage assembly. The single rolling element complement bore diameter F_{ws} and the outside diameter E_{ws} can not be measured directly.

3. The outside diameter measuring principle

(1) The single outside diameter measuring

The single outside diameter measuring method is shown in Table 1 – 57.

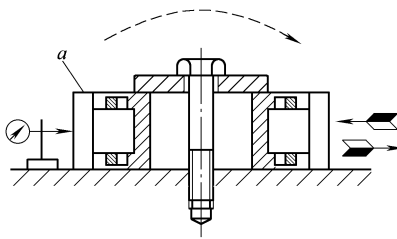
Table 1 – 57 The single outside diameter measuring method

method	illustration
a — measuring region Calibrate the instrument to zero by the proper gauge or the standard ring. Within the stipulate measuring region, in a single radial plane and in several angle directions, measure and record the maximum and the minimum outside diameters $D_{s\max}$ and $D_{s\min}$. Within several radial planes, repeat measuring and recording the results, then determine the maximum and minimum single ring outside diameters $D_{s\max}$ and $D_{s\min}$.	This method can apply to all kinds of rolling bearing ring, shaft washer and housing washer. The single outside diameter D_{sp} or D_s could be read directly from the indicator. In order to avoid the influence of the gravitational force, the axis of the bearing ring or washer should be on the vertical position. According to the measuring values of $D_{s\max}$ and $D_{s\min}$, we could get the following values: D_m — the single plane mean outside diameter; Δ_{Dm} — the deviation of the single plane mean outside diameter; V_{Dm} — the mean outside diameter variation. According to the measuring values of D_s, $D_{s\max}$ and $D_{s\min}$, we could get the following values: D_m — mean outside diameter; Δ_{Dm} — mean outside diameter deviation; Δ_{Ds} — single outside diameter deviation; V_{Ds} — outside diameter variation.

(2) The single outside diameter measuring of the rolling element complement

The single outside diameter measuring method of the rolling element complement is shown in Table 1 – 58.

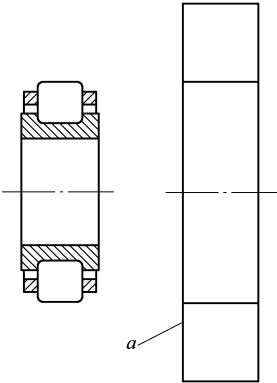
Table 1 – 58 The single outside diameter measuring of the rolling element complement

method	illustration																					
 <i>a</i> — round gauge Fix the inner ring of the bearing without outer ring on to the level platform. Slip the round gauge onto the outside diameter of the rolling element complement. Put the indicator instrument onto the round gauge outside surface perpendicular to the middle position of the inner ring width. In the same radial direction of the indicator, apply the enough reciprocating radial load on to the round gauge and measure the radial movement of the round gauge. The applied radial load value is shown in the table. Record the reading at the radial limit position of the round gauge. Repeat measuring on several angle positions, to determine the largest and the smallest readings of E_{wsmax} and E_{wsmin}.	This method can apply to the cylindrical roller bearing without outer ring and the needle roller bearing without the outer ring. The complement single outside diameter E_{ws} equals to the round gauge inner ring bore diameter minus the measuring value. According to the E_{wsmax} and the E_{wsmin}, we can get: E_{wm} — rolling element complement mean outside diameter; Δ_{Ewm} —the deviation of the rolling element complement mean outside diameter. <table><tr><th colspan="3">Radial measuring load</th></tr><tr><th colspan="2">E_w/mm</th><th>measuring load/N</th></tr><tr><th>over</th><th>incl.</th><th>min</th></tr><tr><td>—</td><td>30</td><td>50</td></tr><tr><td>30</td><td>50</td><td>60</td></tr><tr><td>50</td><td>80</td><td>70</td></tr><tr><td>80</td><td>—</td><td>80</td></tr></table>	Radial measuring load			E_w/mm		measuring load/N	over	incl.	min	—	30	50	30	50	60	50	80	70	80	—	80
Radial measuring load																						
E_w/mm		measuring load/N																				
over	incl.	min																				
—	30	50																				
30	50	60																				
50	80	70																				
80	—	80																				

(3) Function inspection of the largest single outside diameter of the rolling element complement

The largest single outside diameter function inspection method of the rolling element complement is shown in Table 1 – 59.

Table 1 – 59 The largest single outside diameter function inspection method of the rolling element complement

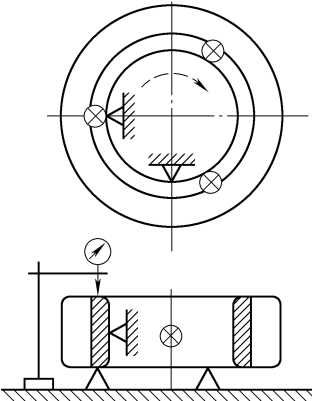
method	illustration
 <i>a</i> — round gauge The rolling element complement outside diameter E_w is measured by the pass end and the stop end of the round gauge. The round gauge dimension is 0.002mm larger than the largest outside diameter of the rolling element complement. The round gauge stop end dimension is 0.002mm smaller than the smallest outside diameter of the rolling element complement.	This method can apply to the bearings without outer ring of the cylindrical roller bearing and the needle roller bearing. The round gauge pass end should pass the rolling element complement. But the round gauge stop end should not pass the rolling element complement. The round gauge is only used for inspecting the dimension limit, but can not be directly used for measuring the single outside diameter E_{ws} of the rolling element complement. This inspecting method could confirm whether the E_{wsmax} range is within the tolerance limit range.

4. The measuring principle of the width and the height

(1) The single width measuring of the bearing ring

The single width measuring method of the bearing ring is shown in Table 1 – 60.

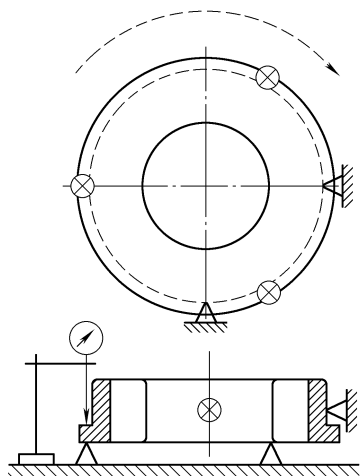
Table 1 – 60 The single width measuring method of the bearing ring

method	illustration
 Calibrate the instrument to zero by the proper gauge or standard ring that departs a proper distance from the base end. Support the one end face of the ring on the three evenly divided fixed supporting points with equal height. The bore surface is supported by two proper radial supporting points that form the angle 90° each other for centering the ring. Put the indicator on the other end face at the right above position of a fixed supporting point. Rotate the ring for a circle, measure and record the largest and smallest single ring widths B_{smax} and B_{smin} (C_{smax} and C_{smin}).	This method can apply to the inner ring and outer ring of all kinds of rolling bearings. The single ring width B_s or C_s is the actual measuring value on any point of the ring. According to the single width B_s or C_s of the inner ring or the outer ring, we can get: Δ_{Bs} or Δ_{Cs} — the single ring width deviation V_{Bs} or V_{Cs} — the ring width variation B_m or C_m — the ring mean width

(2) Measuring of the single width of the outer ring flange

The measuring method of the single width of the outer ring flange is shown in Table 1 – 61.

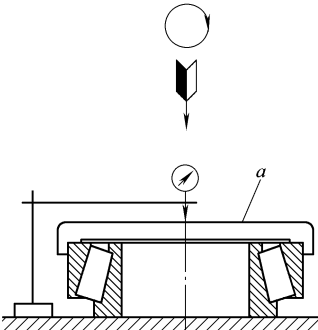
Table 1 – 61 The measuring method of the single width of the outer ring flange

method	illustration
 Calibrate the instrument to zero by the proper gauge or the standard ring that departs a proper height from the fixed supporting point. Support the outer ring flange front face on the three evenly divided fixed supporting points with equal height, The bearing outside surface is supported by two proper radial supporting points that form the angle 90° each other for centering the outer ring. Put the indicator on the after flange back face at the right up position of a fixed supporting point. Rotate the outer ring for a circle, measure and record the largest and smallest single widths C_{1smax} and C_{1smin} of the outer ring flange.	This method can apply to all kinds of the radial rolling bearings with the outer ring flange. The single outer ring flange width C_{1s} is the actual measuring value on any point of the flange back face. According to the single width C_{1s} of the outer ring flange, can get: Δ_{1s} — the single outer ring flange width deviation V_{C1s} — the outer ring flange width variation

(3) The main measuring method of actual bearing width

The main measuring method of actual bearing width is shown in Table 1 – 62.

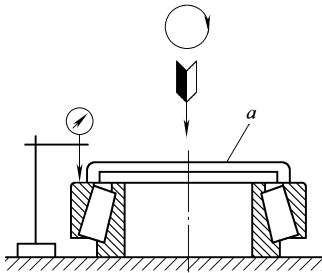
Table 1 – 62 The main measuring method of actual bearing width

method	illustration
 a — plate Calibrate the instrument to zero by the proper gauge or the standard ring that departs a proper height from the fixed supporting point. Support the base end face of the inner ring. Guarantee the really contacting of the rolling elements with the raceway. For the tapered roller bearing, we should also guarantee the contacting of the rolling elements with the back face rib of the inner ring and raceway. Put a level plate of known height onto the outer ring base end face. Apply a stable central axial load, and let the indicator point on the center of the level plate. Rotate the outer ring several times, let the width arriving at its smallest value, and then read the reading.	This method is the main method for measuring the actual width of the radial and the angular contact bearing, which width is confined by one inner ring end face and one outer ring end face. This measuring method does not contain the influence of the flatness of the ring end face. The actual bearing width T_s equals to the indicator reading minus the known plate height. The actual bearing width deviation Δ_{T_s} can be obtained from the measuring value of T_s.

(4) Another measuring method of the actual bearing width

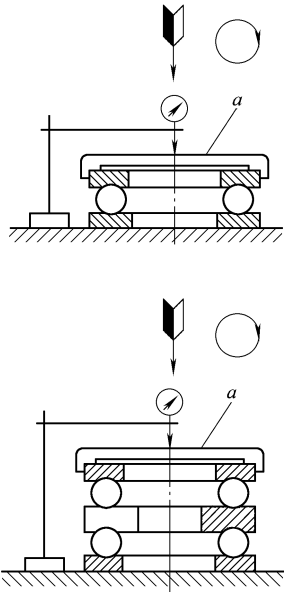
Another measuring method of the actual bearing width is shown in Table 1 – 63.

Table 1 – 63 Another measuring method of the actual bearing width

method	illustration
 a — stable plate Calibrate the instrument to zero by the proper gauge that departs a proper height from the fixed supporting point or by the standard ring. Support the base end face of the inner ring. Guarantee the really contacting of the rolling elements with the raceway. For the tapered roller bearing, we should also guarantee the contacting of the rolling elements with the back face rib of the inner ring and raceway. Put a stable level plate onto the outer ring base end face. Apply a stable central axial load. Put the indicator on the outer ring base end face. Rotate the outer ring and read the readings. On the several positions of the outer ring back circle direction and radial direction repeat to read the readings, to determine the actual bearing width T_s value.	This method is the main method for measuring the actual width of the radial and the angular contact bearing, whose width is confined by one inner ring end face and one outer ring end face. This measuring method applies to the tapered roller bearing, the single row spherical roller bearing, the single row angular contact ball bearing, and the thrust self-aligning roller bearing. The actual bearing width deviation Δ_{T_s} can be obtained from the measuring value of T_s. This measuring method is another method of measuring actual bearing width T_s. The actual bearing width T_s is the arithmetic average value of the indicator readings. There is no need to use the stable level plate or ring for large type bearings. This measuring method contains the influence of the flatness of the ring end face.

(5) The measurement of the actual thrust bearing height The measurement of the actual thrust bearing height is shown in Table 1 – 64.

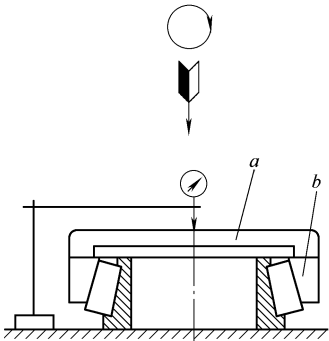
Table 1 – 64 The measurement of the actual thrust bearing height

method	illustration
 <i>a</i> — plate Support the bearing onto a level plate. Calibrate the instrument to zero by a proper gauge that departs a proper height from the plate or the standard ring bearing. Apply a stable central axial load. Put a known height level plate onto the assembled bearing, and put the indicator on the center of the level plate. Rotate the bearing several times, and must let the bearing arrive at its smallest height, then read the reading.	The actual bearing height T_s equals to the indicator reading minus the known plate height. This measuring method does not contain the influence of the flatness of the washer end face. The actual bearing height deviation Δ_{Ts} can be obtained from the measuring value of T_s.

(6) The measurement of the actual effective width of the tapered roller bearing inner assembly

The measurement of the actual effective width of the tapered roller bearing inner assembly is shown in Table 1 – 65.

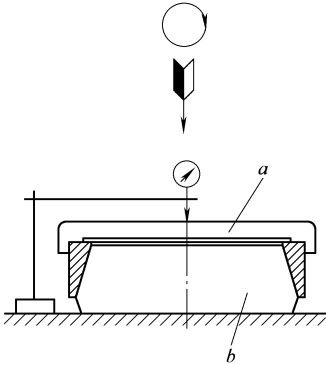
Table 1 – 65 The measuring of the actual effective width of the tapered bearing inner assembly

method	illustration
 <i>a</i> — level plate <i>b</i> — standard outer ring Calibrate the instrument to zero by a proper gauge that departs a proper height from the plate or the standard ring. Support the inner ring base end face of the inner assembly. Guarantee the contact of the rolling elements with the inner ring back rib and the raceway. Put the standard outer ring onto the inner assembly. Put a known height level plate onto the standard outer ring back face, apply a stable central axial load, and put the indicator on the center of the level plate. Rotate the outer ring several times, and must let it arrive at its smallest width, and then read the reading.	This method applies to the tapered roller bearing inner assembly, and need use the standard outer ring. The actual effective width T_{1s} is based on the standard outer ring height, and equals to the indicator reading minus the known level plate height. This measuring method does not contain the influence of the flatness of the ring end face.

(7) The measurement of the actual effective width of the tapered roller bearing outer ring

The measurement of the actual effective width of the tapered roller bearing outer ring is shown in Table 1 - 66.

Table 1 - 66 The measurement of the actual effective width of the tapered bearing outer ring

method	illustration
 <i>a</i> — level plate <i>b</i> — inner standard plug gauge Calibrate the instrument to zero by a proper gauge that departs proper height from the plate or the standard ring. Support the inner standard plug gauge back face on a level plate, put the outer ring on the plug gauge. Put a knowing height plate on the outer ring back face, applies a stable central axial load on it, and put the indicator on the plate center. Rotate the outer ring several times, and must let it arrives its smallest width, then read the reading.	This method can apply to the tapered roller bearing outer ring, and need use the standard inner plug gauge. The actual effective width T_{2s} is based on the standard inner plug gauge height, and equals to the indicator reading minus the known level plate height. This measuring method does not contain the influence of the flatness of the ring end face. If there is a need, we can use the corrected inner assembly (inner ring, cage and rolling element subunit) and substitutes for the standard inner plug gauge.

5. The measuring principle of the chamfer of a ring and a washer

(1) The single chamfer dimension measuring

The main measuring method of the single chamfer dimension is shown in Table 1 – 67.

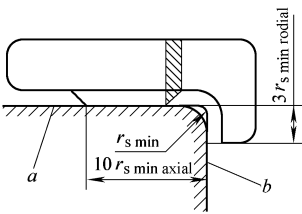
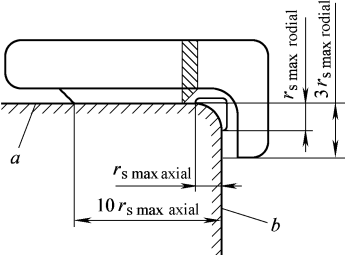
Table 1 – 67 The single chamfer dimension measuring method

method	illustration
a — bore or outer diameter surface b — end face Draw out the chamfer section contour by magnifying at least 20 * times, extending the diameter surface to the intersecting point with the end face contour base line, measuring the horizontal and the vertical distances from the intersecting point to the beginning point of the diameter surface and the end face. Draw out the round arc of a radius r_{smin}. If the axial direction dimension is different from the radial direction dimension, we can use the smaller one of the two chamfer dimensions.	This measuring method of radius r_s can apply to all types of bearing inner ring, outer ring and thrust washer. The bearing ring chamfer should not go beyond the round arc of radius r_{smin}. Note: 1. The limit value of r_{smax} could be different for the axial direction and the radial direction. 2. This method also applies to the measuring of the appointed radius r_1 and r_2.

(2) Function measurement of the single chamfer dimension

The function measurement method of the single chamfer dimension is shown in Table 1 – 68.

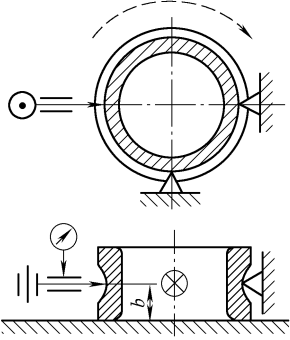
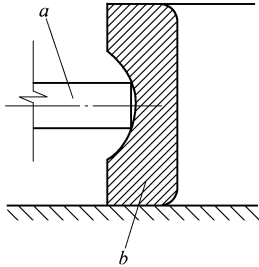
Table 1 – 68 The function measuring method of the single chamfer dimension

method	illustration
 a — bore or outer ring surface b — end face Put the smallest chamfer sample plate onto the ring or washer. The sample plate should lean reliably against the diameter surface and the end face. Compare the chamfers of ring or washer with the contour of the sample plate.	The inspecting method for the radius r_s applies to all types of bearing inner ring, outer ring and thrust washer. The ring or washer chamfer should not interfere with the smallest chamfer (r_{smin}) sample plate. The bearing ring or washer chamfer should not go beyond the marking sign on the largest chamfer (r_{smax}) sample plate. Note: 1. The limit value of r_{smax} could be different for the axial direction and the radial direction. 2. This method also applies to the inspecting of the appoint r_1 and r_2.
 a — bore or outer ring surface b — end face Put the largest chamfer sample plate onto the ring or the washer. The sample plate should lean reliably against the diameter surface and the end face. Compare the chamfers of ring or washer with the marking sign of the sample plate.	

6. The measuring method of raceway parallelism

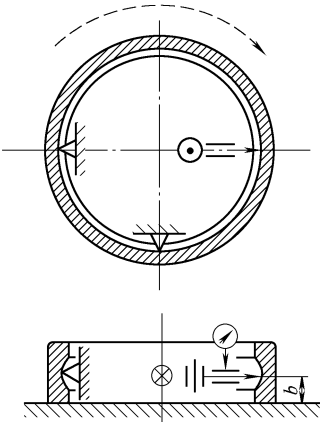
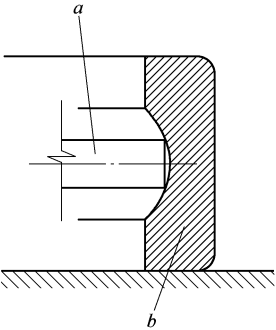
(1) The measurement of the inner raceway parallelism with respect to the face The measuring method of the inner raceway parallelism with respect to the face is shown in Table 1 – 69.

Table 1 – 69 The measuring method of the inner raceway parallelism with respect to the face

method	illustration
 Support the inner ring base face on a level plate. On the middle part of the raceway, support the raceway surface by two supporting points that form a 90° angle, for centering the inner ring. The measuring tip points exactly to one fixed supporting point, and guarantee that the measuring tip is pressing on the raceway with a constant pressure. The pressure direction is parallel with the ring axial line. Rotate the inner ring for one circle, read the instrument indicator reading.	This method applies to all of the radial ball bearings. The inner ring raceway parallelism S_1 with respect to the face is the difference between the largest reading and smallest reading of the instrument indicator. The measuring tip height b is located on the raceway contact diameter place. In actual measuring, we can use the measuring tip that has the same curvature as the raceway for improving the measuring tip axial direction swing (see the following figure).  a — measuring tip b — inner ring

(2) The measurement of the outer raceway parallelism with respect to the face The measuring method of the outer raceway parallelism with respect to the face is shown in Table 1 – 70.

Table 1 – 70 The measuring method of the outer raceway parallelism with respect to the face

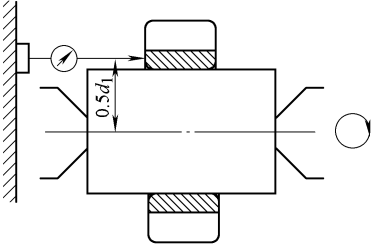
method	illustration
 Support the outer ring base face on a level plate. On the middle part of the raceway, support the raceway surface by two supporting points that form a 90° angle, for centering the outer ring. The measuring tip points exactly to one fixed supporting point, and guarantee that the measuring tip is pressing on the raceway with a constant pressure. The pressure direction is parallel with the ring axial line. Rotate the outer ring for one circle, and read the instrument indicator reading.	This method applies to all of the radial ball bearings. S_e is the difference of the largest reading and smallest reading of the instrument indicator, which is the outer ring raceway parallelism with respect to the face. The measuring tip height b is located on the raceway contact diameter place. In actual measuring, we can use the measuring tip that has the same curvature as the raceway for improving the measuring tip axial direction swing (see the following figure).  a — measuring tip b — outer ring

7. The measuring method of the axial runout of the inner ring face with respect to the bore

(1) The measuring method A of the axial runout of the inner ring face with respect to the bore

The measuring method A of the axial runout of the inner ring face with respect to the bore is shown in Table 1-71.

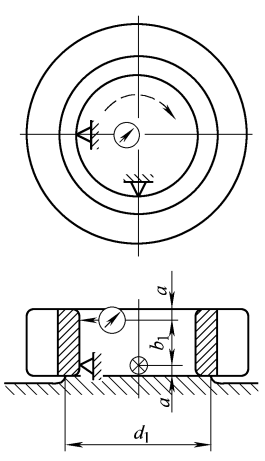
Table 1-71 The measuring method A of the axial runout of the inner ring face with respect to the bore

method	illustration
 Use the precise mandrel of taper as 1:5000. Mount the assembled bearing onto the tapered mandrel, and install the mandrel within the two tip-tops for guaranteeing its precise rotating. Put the instrument indicator on such position of the inner base face that the radial distance from the mandrel axis is the 1/2 mean diameter of the inner base face. Rotate the inner ring for one circle, and read the reading of the instrument indicator.	This method applies to radial bearing and its inner ring, and mostly applies to the inner ring of which the proportion of the inner diameter and the width is smaller than 4. The axial runout S_d of the inner ring face with respect to the bore equals to the difference of the largest reading and the smallest reading of the instrument indicator. Note: while mounting the bearing onto the mandrel, we must be very carefully, and let the inner ring axis and the mandrel axis in co-axial line. d_1 — inner ring face mean diameter

(2) The measuring method B of the axial runout of the inner ring face with respect to the bore

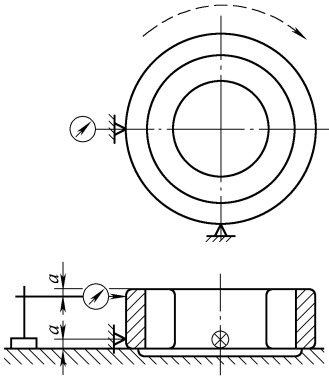
The measuring method B of the axial runout of the inner ring face with respect to the bore is shown in Table 1 – 72.

Table 1 – 72 The measuring method B of the axial runout of the inner ring face with respect to the bore

method	illustration
 Support the inner ring base face on a level plate. If it is an assembled bearing, we should let the outer ring in free condition, and support the inner ring bore surface by two supporting points that form a 90° angle, for centering the inner ring. The instrument indicator should place on the exact above position of one supporting point. The instrument indicator and the two supporting points are separably placed on the limited positions of the chamfer measuring region. Rotate the inner ring for one circle, and read the reading of the instrument indicator.	This method applies to all types of the radial bearing and its inner ring, and most applies to the large ring or the inner ring whose proportion of the inner diameter and the width is not smaller than 4. In that case, the bearing weight would influence the measuring result. The measuring result of this method is the axial runout of the bore with respect to the inner ring face. Through calculation we can convert it as the axial runout S_d of the inner ring face with respect to the bore: $S_d = \frac{S_{dr} * d_1}{2 * b_1}$ In the equation: S_d — the axial runout of the inner ring face with respect to the bore S_{dr} — the difference of the largest reading and the smallest reading of the indicator d_1 — the inner ring face mean diameter b_1 — the axial distance between the indicator and its exact below supporting point

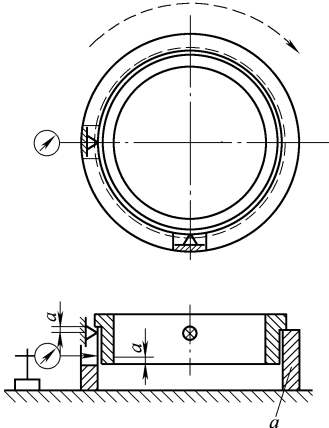
(3) The axial runout measuring of the outer ring outside surface with respect to the face The axial runout measuring of the outer ring outside surface with respect to the face is shown in Table 1 – 73.

Table 1 – 73 The axial runout measuring method of the outer ring outside surface with respect to the face

method	illustration
	This method applies to all types of the radial bearing and its outer ring, and most applies to the large ring or the outer ring whose proportion of the inner diameter and the width is not smaller than 4. In that case, the bearing weight would influence the measuring result. The axial runout S_D of the outer ring outside surface respect to the face equals to the difference of the largest reading and the smallest reading of the indicator.

(4) The axial runout measuring of the outer ring outside surface with respect to the flange back face The axial runout measuring method of the outer ring outside surface with respect to the flange back face is in Table 1 – 74.

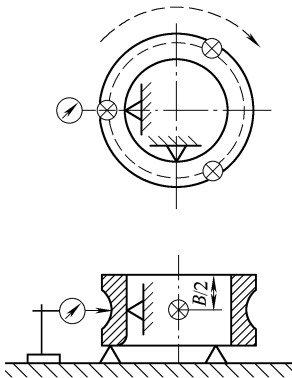
Table 1 – 74 The axial runout measuring method of the outer ring outside surface with respect to the face

method	illustration
 <i>a</i> — cylindrical supporting round ring Support the outer ring flange back face on a cylindrical supporting round ring. If there is an assembled bearing, we should let the inner ring in free condition. The supporting round ring inner diameter equals to the flange mean diameter. Support the outer ring outside surface by two supporting points that form a 90° angle each other, for centering the outer ring. Note: The slot on the supporting round ring allows the entering of the side supporting point. Place the indicator on the exact lower position of a supporting point. The instrument indicator and the two supporting points are separably placed on the limited positions of the chamfer measuring region. Rotate the outer ring for one circle, and read the readings of the instrument indicator.	This method can apply to all types of the flange outer ring radial bearing. The axial runout of the outer ring outside surface with respect to the flange back face S_{DI} equals to the difference of the largest reading and the smallest reading of the indicator.

8. The measuring method of the variation in thickness

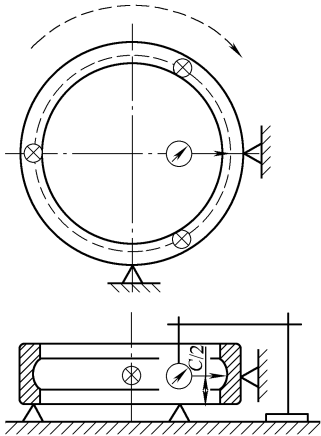
(1) The measuring method of the variation in thickness between the inner ring raceway and the bore The measuring method of the variation in thickness between the inner ring raceway and the bore is shown in Table 1 – 75.

**Table 1 – 75 The measuring method of the variation in thickness
between the inner ring raceway and the bore**

method	illustration
 Support an inner ring end face on three evenly distributed, equal height fixed supporting points. Support the bore surface by two proper radial supporting points that form a 90° angle each other. The distance departing from the end face is $B/2$ or exactly on the raceway middle part, for centering the inner ring. The instrument indicator points exactly to one bore supporting point. Rotate the inner ring for one circle, and read the readings of the instrument indicator.	This method applies to all types of the radial and angular contact bearing inner ring. The variation in thickness between the inner ring raceway and bore K_i equals to the difference of the largest reading and the smallest reading of the indicator.

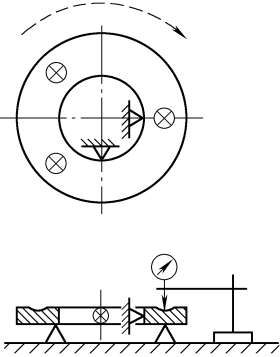
(2) The measurement of the variation in thickness between outer ring raceway and outside surface The measuring method of the variation in thickness between outer ring raceway and outside surface is shown in Table 1 – 76.

**Table 1 – 76 The measuring method of the variation in thickness between
outer ring raceway and outside surface**

method	illustration
 Support an outer ring end face on three evenly distributed, equal height fixed supporting points. Support the outside surface by two proper radial supporting points that form 90° angle each other. The distance departing from the end face is $C/2$ or exactly on the raceway middle part, for centering the outer ring. The instrument indicator points exactly to one outer diameter supporting point. Rotate the outer ring for one circle, and read the readings of the instrument indicator.	This method can apply to all types of the radial and angular contact bearing outer ring. The variation in thickness between outer ring raceway and outside surface K_e equals to the difference of the largest reading and the smallest reading of the indicator.

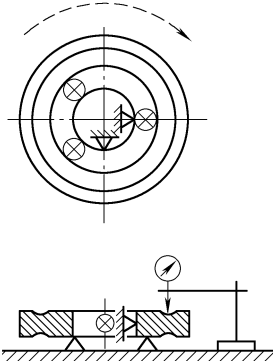
(3) The measuring of the variation in thickness between shaft washer raceway and back face The measuring method of the variation in thickness between shaft washer raceway and back face is shown in Table 1 – 77.

Table 1 – 77 The measuring method of the variation in thickness between shaft washer raceway and back face

method	illustration
 Support a shaft washer end face on three evenly distributed, equal height fixed supporting points. Support the inner surface by two proper radial supporting points that form a 90° angle each other for centering the shaft washer. The instrument indicator is placed exactly on the raceway middle part. The instrument indicator points exactly to the above direction of one shaft washer diameter supporting point. Rotate the shaft washer for one circle, and read the readings of the instrument indicator.	This method applies to all types of shaft washer having planar or shaped raceway and having flat end face. The variation in thickness between shaft washer raceway and back surface S_1 equals to the difference of the largest reading and the smallest reading of the indicator.

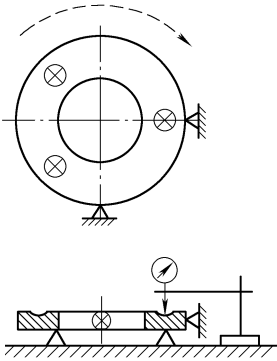
(4) The measuring of the variation in thickness between central ring raceway and back face The measuring method of the variation in thickness between central raceway and back face is shown in Table 1 – 78.

**Table 1 – 78 The measuring method of the variation in thickness
between central ring raceway and back face**

method	illustration
 Support a center ring end face on three evenly distributed, equal height fixed supporting points. Support the bore surface by two proper radial supporting points that form a 90° angle each other, for centering the center ring. The instrument indicator is placed on the middle part of the raceway, opposite against one supporting point. The center ring contacts with the supporting points, rotate the center ring for one circle, and read the readings of the instrument indicator. Repeat the measuring for the other raceway.	This method applies to the center ring that has the shaped raceway on every end face. The variation in thickness between center ring raceway and back face S_i equals to the difference of the largest reading and the smallest reading of the indicator. The variation in thickness between every back face and raceway is measured independently.

(5) The measuring of the variation in thickness between housing washer raceway and back face The measuring method of the variation in thickness between housing washer raceway and back face is shown in Table 1 – 79.

Table 1 – 79 The measuring method of the variation in thickness between housing washer raceway and back face

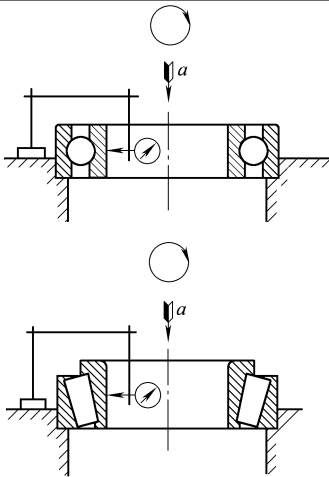
method	illustration
 Support the housing washer flat face on three evenly distributed, equal height fixed supporting points. Support the outside surface by two proper radial supporting points that form a 90° angle each other, for centering the housing washer. The instrument indicator is placed on the middle part of the raceway and points to the exact above direction of one supporting point. The housing washer contacts with the supporting points, rotate the housing washer for one circle, and read the readings of the instrument indicator.	This method can apply to the housing washer that has the flat raceway or shaped raceway and flat end face. The variation in thickness between housing washer raceway and back face S_e equals to the difference of the largest reading and the smallest reading of the indicator.

9. The radial runout measuring method

(1) The main measuring method of the radial runout of an assembled bearing inner ring

The main measuring method of the radial runout of an assembled bearing inner ring is shown in Table 1 – 80.

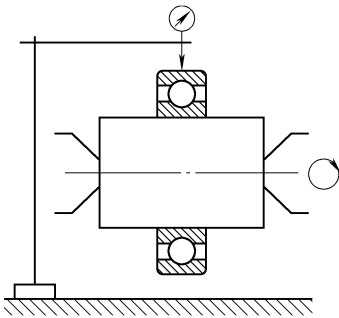
Table 1 – 80 The main measuring method of the radial runout of an assembled bearing inner ring

method	illustration
 a — the load on inner ring Support the outer ring base face on a level plate with a guider, for centering the ring outer diameter. Apply a stable central axial load on the inner ring base face. Guarantee the contact of rolling elements with the raceway. For the tapered roller bearing, we should also guarantee the contact of rolling element with the inner back face rib and the raceway. The instrument indicator is placed on the inner ring bore surface, as close as possible to the middle part of the inner ring raceway. Rotate the inner ring for one circle, and read the readings of the instrument indicator.	This method applies to radial ball bearing (include single row angular contact ball bearing), four-point contact ball bearing and tapered roller bearing. The radial runout K_{ia} of an assembled bearing inner ring equals to the difference of the largest reading and the smallest reading of the indicator. There are many factors to influence the measuring result for assembled bearing inner ring radial runout (such as: the variation of the rolling element diameter, raceway defect and waviness, contact angle variation, base face/surface flatness and lubricant impurity). It is very difficult to measure precisely, especially in higher precision bearing case. If there is a controversy between the bearing maker and the customer, they can discuss to decide the effective right measuring method.

(2) Another measuring method of the radial runout of an assembled bearing inner ring

Another measuring method of the radial runout of an assembled bearing inner ring is shown in Table 1 – 81.

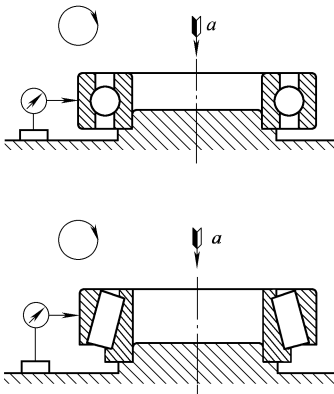
Table 1 – 81 Another measuring method of the radial runout of an assembled bearing inner ring

method	illustration
 Use precision mandrel of taper as 1:5000. Mount an assembled bearing onto the taper mandrel, and mount the mandrel on between two tip-tops, and guarantee its precise rotating. The instrument indicator is placed on the outer ring outside surface, as close as possible to the middle part of the outer ring raceway. Keep the outer ring on static position, and guarantee that its weight is supported by the rolling elements. Rotate the mandrel for one circle, and read the readings of the instrument indicator.	This method applies to radial ball bearing (except the single row angular contact ball bearing), cylindrical roller bearing, self-aligning roller bearing and needle roller bearing. The assembled bearing inner ring radial runout K_{ia} equals to the difference of the largest reading and the smallest reading of the indicator. There are many factors to influence the measuring result for the assembled bearing inner ring radial runout (such as: the variation of the rolling element diameter, raceway defect and waviness, contact angle variation, base face/surface flatness and lubricant impurity). It is very difficult to measure precisely, especially in higher precision bearing case. If there is a controversy between the bearing manufacturer and the customer, they can discuss to decide the effective right measuring method.

(3) The main measuring method of the radial runout of an assembled bearing outer ring

The main measuring method of the radial runout of an assembled bearing outer ring is shown in Table 1 – 82.

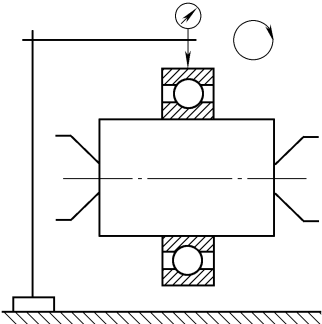
Table 1 – 82 The main measuring method of the radial runout of an assembled bearing outer ring

method	illustration
 a — the load on outer ring Support the inner ring base face on a level plate with a guider, for centering the ring bore. Apply a stable central axial load on the outer ring base face. Guarantee the contact of rolling elements with the raceway. For the tapered roller bearing, we should also guarantee the contact of rolling elements with the inner ring back face rib and the raceway. The instrument indicator is placed on the outer ring outside surface, as close as possible to the middle part of the outer ring raceway. Rotate the ring for one circle, and read the readings of the instrument indicator.	This method applies to radial ball bearing (include single row angular contact ball bearing), four-point contact ball bearing and tapered roller bearing. The assembled bearing inner ring radial runout K_{ca} equals to the difference of the largest reading and the smallest reading of the indicator. There are many factors to influence the measuring result for the assembled bearing outer ring radial runout (such as: the variation of the rolling element diameter, raceway defect and waviness, contact angle variation, base face/surface flatness and lubricant impurity). It is very difficult to measure precisely, especially in higher precision bearing case. If there is a controversy between the bearing maker and the customer, they can discuss to decide the effective right measuring method.

(4) Another measuring method of the radial runout of an assembled bearing outer ring

Another measuring method of the radial runout of an assembled bearing outer ring is shown in Table 1 – 83.

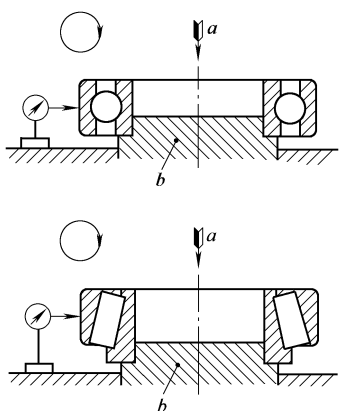
Table 1 – 83 Another measuring method of the radial runout of an assembled bearing outer ring

method	illustration
 Use the precision mandrel of taper 1:5000. Mount the assembled bearing onto the taper mandrel, and mount the mandrel on between two tip-tops for precise rotating. The instrument indicator is placed on the outer ring outside surface as close as possible to the middle part of the outer ring raceway. Keep the inner ring in static state. Rotate the outer ring for one circle, and read the readings of the instrument indicator.	This method applies to radial ball bearing (except the single row angular contact ball bearing), cylindrical roller bearing and needle roller bearing. The assembled bearing outer ring radial runout K_{ca} equals to the difference of the largest reading and the smallest reading of the indicator. There are many factors to influence the measuring result for the assembled bearing outer ring radial runout (such as: the variation of the rolling element diameter, raceway defect and waviness, contact angle variation, base face/surface flatness and lubricant impurity). It is very difficult to measure precisely, especially in higher precision bearing case. If there is a controversy between the bearing maker and the customer, they can consult to select the effective right measuring method.

(5) The measurement of the radial runout of an assembled bearing inner ring

The measurement of the radial runout of an assembled bearing inner ring is shown in Table 1 – 84.

Table 1 – 84 The measurement of the radial runout of an assembled bearing inner ring

method	illustration
 a — the load on inner ring b — rotating plate Support the inner ring base face on a rotating plate with a guider, for centering the inner ring diameter. Apply a stable central axial load on the outer ring base face, to guarantee the contact of rolling elements with the raceway. For the tapered roller bearing, we should also guarantee the contact of rolling elements with the inner ring back face rib and the raceway.	This method applies to radial ball bearing (include single row angular contact ball bearing), four-point contact ball bearing and tapered roller bearing. The assembled bearing inner ring asynchronous radial runout K_{iaa} equals to the maximum reading of the instrument indicator measured when the inner ring rotating many circles and on different outer ring fixed points. When measuring, the inner ring should be rotating many times in both clockwise and anti-clockwise directions.

(continued)

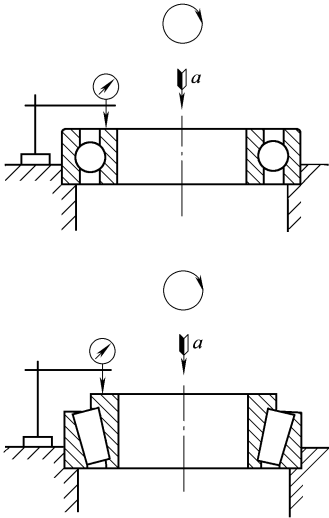
method	illustration
The instrument indicator is placed on the static outer ring outside surface , and should be placed as close as possible to the middle part of the outer ring raceway. Rotate the inner ring for seven circles in both directions. Record the maximum reading of the instrument indicator in every circle. The instrument indicator is placed on another radial position of the outer ring outside surface. The inner ring rotates in both directions for many times. Repeat measuring. There is no relative rotating between the bearing inner ring and the rotating plate. The instrument indicator is placed on different radial positions of the outer ring outside surface. Repeat measuring.	There are many factors to influence the measuring result for the assembled bearing inner ring radial runout (such as: the variation of the rolling element diameter, raceway defect and waviness, contact angle variation, base face/surface flatness and lubricant impurity). It is very difficult to measure precisely, especially in higher precision bearing case. If there is a controversy between the bearing maker and the customer, they can consult to select the effective right measuring method.

10. The axial runout measuring method

(1) The measurement of the axial runout of an assembled bearing inner ring

The measurement of the axial runout of an assembled bearing inner ring is shown in Table 1 – 85.

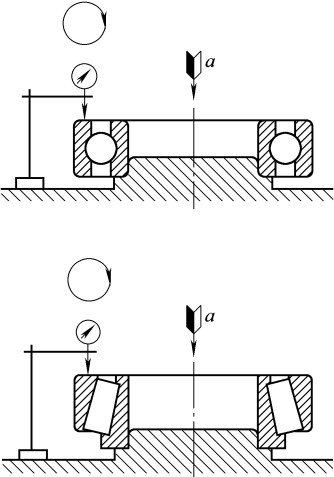
Table 1 – 85 The measurement of the axial runout of an assembled bearing inner ring

method	illustration
 a — the load on inner ring Support the outer ring base face on a level plate with a guider, for centering the ring outside diameter. Apply a stable central axial load on the inner ring base face, to guarantee the contact of rolling elements with the raceway. For the tapered roller bearing, we should also guarantee the contact of rolling elements with the inner ring back face rib and the raceway. The instrument indicator is placed on the inner ring base face. Rotate the inner ring for one circle, and read the readings of the instrument indicator.	This method applies to radial ball bearing (include single row angular contact ball bearing), four-point contact ball bearing and tapered roller bearing. The assembled bearing inner ring axial runout S_{ia} equals to the difference of the largest reading and the smallest reading of the indicator. There are many factors to influence the measuring result for the assembled bearing inner ring axial runout (such as: the variation of the rolling element diameter, raceway defect and waviness, contact angle variation, base face/surface flatness and lubricant impurity). It is very difficult to measure precisely, especially in higher precision bearing case. If there is a controversy between the bearing maker and the customer, they can consult to select the effective right measuring method.

(2) The measurement of the axial runout of an assembled bearing outer ring

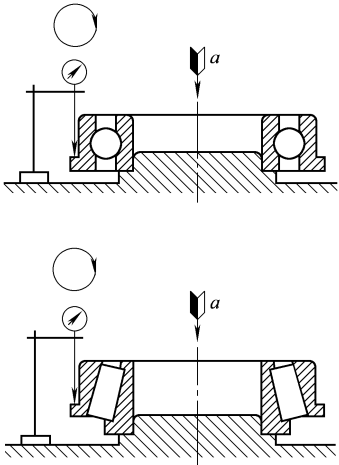
The measurement of the axial runout of an assembled bearing outer ring is shown in Table 1 – 86.

Table 1 – 86 The measurement of the axial runout of an assembled bearing outer ring

method	illustration
 a — the load on outer ring Support the inner ring base face on a level plate with a guider, for centering the ring bore. Apply a stable central axial load on the outer ring base face, to guarantee the contact of rolling elements with the raceway. For the tapered roller bearing, we should also guarantee the contact of rolling elements with the inner ring back face rib and the raceway. The instrument indicator is placed on the outer ring base face. Rotate the outer ring for one circle, and read the readings of the instrument indicator.	This method applies to radial ball bearing (include single angular contact ball bearing), four-point contact ball bearing and tapered roller bearing. The assembled bearing outer ring axial runout S_{ca} equals to the difference of the largest reading and the smallest reading of the indicator. There are many factors to influence the measuring result for the assembled bearing outer ring axial runout (such as: the variation of the rolling element diameter, raceway defect and waviness, contact angle variation, base face/surface flatness and lubricant impurity). It is very difficult to measure precisely, especially in higher precision bearing case. If there is a controversy between the bearing maker and the customer, they can consult to select the effective right measuring method.

(3) The measurement of the axial runout of an assembled bearing outer ring flange back face The measurement of the axial runout of an assemble bearing outer ring flange back face is shown in Table 1 – 87.

Table 1 – 87 The measurement of the axial runout of an assembled bearing outer ring flange back face

method	illustration
 a — the load on outer ring Support the inner ring base face on a level plate with a guider, for centering the inner ring bore. Apply a stable central axial load on the outer ring base face, to guarantee the contact of rolling elements with the raceway. For the tapered roller bearing, should also guarantee the contact of rolling elements with the inner ring back face rib and the raceway. The instrument indicator is placed on the outer ring flange back face and on the middle part of the flange. Rotate the outer ring for one circle, and read the readings of the instrument indicator.	This method can apply to radial ball bearing with outer ring flange (include single row angular contact ball bearing), four-point contact ball bearing and tapered roller bearing. The assembled bearing outer ring flange back face axial runout S_{cal} equals to the difference of the largest reading and the smallest reading of the indicator. There are many factors to influence the measuring result for the assembled bearing outer ring flange back face axial runout (such as: the variation of the rolling element diameter, raceway defect and waviness, contact angle variation, base face/surface flatness and lubricant impurity). It is very difficult to measure precisely, especially in higher precision bearing case. If there is a controversy between the bearing maker and the customer, they can consult to select the effective right measuring method.

1.5 Materials for Rolling Bearings

1.5.1 Requirements for Rolling Bearing Materials

Because rolling bearing usually runs under high contact stress and high speed, first of all, the materials used for making rolling bearing ring and rolling element should have high strength and hardness and good through-hardening-ability performance; secondly, the materials should have good fatigue strength and wear resistance for having enough long life. Besides, in order to keep normal running under working temperature, bearing materials must have dimension stability and toughness.

The functional performance and endurance of a dimensionally perfect bearing with ideal internal geometries and surfaces, correct mounting, and preferential operating conditions are significantly influenced by the characteristics of its materials and its metallurgical processes. Since the beginning of the 20th century, the High-Carbon-Chromium bearing steel has been introduced in making rolling bearings and it is still the mostly-used steel for ball bearings and for many bearing applications, whose mass ratio is $\omega_C < 0.8$.

Along with the continuous development of science and technology, the demand for rolling bearing performance has consistently increased, such as the high reliability under larger loads and so on. Many new bearing materials have been developed for special working condition bearings (such as under low-temperature freezing condition, vacuum condition, corrosive medium condition).

For large bearing sizes and extra-large bearings, in order to keep their high surface strength and core toughness, recently there has been developed carburizing steel in which the nickel, manganese, and molybdenum are introduced for increasing the through hardening ability and its mass ratio is $\omega_C < 0.8$.

1. 5. 2 Generally Used Rolling Bearing Steel — High Carbon-Chromium Steel

The main compositions of the High-Carbon-Chromium Bearing Steel are the conventional GCr15 and GCr15SiMn. Recently there has been developed the high hardening bearing steel GCr15SiMo and GCr18Mo and the bearing steel GCr4 with limited hardening ability. The GCr15SiMo and GCr18Mo steel are mainly used for making large diameter bearings that require the entire component having through-hardening or bainite hardening. Through surface induction hardening, the GCr4 steel obtains high impact resistance and both high strength and high toughness, which can be used for making railway bearings and rolling mill bearings.

The applications of the High-Carbon-Chromium Bearing Steel are shown in Table 1 – 88.

Table 1 – 88 High-Carbon-Chromium Bearing Steel applications

Material	Main applications
GCr4	Surface hardening bearing components
GCr15	Most commonly used for rolling bearing rings and rolling elements for bearing rings of wall thickness ≤15mm and rolling element diameter ≤ ϕ25mm
GCr15SiMn	Most commonly used for large rolling bearing rings and rolling elements for bearing rings of wall thickness ≤35mm and rolling element diameter ≤ ϕ50mm
GCr15SiMo	Extra-large rolling bearing rings and rolling elements
GCr18Mo	Wall thickness ≤75mm and rolling element diameter ≤ ϕ90mm

At present, the grade of the High-Carbon-Chromium Bearing Steel made in China and their chemical compositions are listed in Table 1 – 89.

Table 1 – 89 High-Carbon-Chromium Bearing Steel (GB/T 18254)

grade	chemical composition (mass ratio, %)				
	C	Si	Mn	P ≤	S ≤
GCr4	0. 95 ~ 1. 05	0. 15 ~ 0. 30	0. 15 ~ 0. 30	0. 025	0. 020
GCr15	0. 95 ~ 1. 05	0. 15 ~ 0. 30	0. 25 ~ 0. 45	0. 025	0. 025

(continued)

grade	chemical composition (mass ratio, %)				
	C	Si	Mn	P≤	S≤
GCr15SiMn	0.95 ~ 1.05	0.45 ~ 0.75	0.95 ~ 1.25	0.025	0.025
GCr15SiMo	0.95 ~ 1.05	0.65 ~ 0.85	0.20 ~ 0.40	0.027	0.020
GCr18Mo	0.95 ~ 1.05	0.20 ~ 0.40	0.25 ~ 0.40	0.025	0.025

grade	chemical composition (mass ratio, %)			
	Cr	Mo	Ni≤	Cu≤
GCr4	0.35 ~ 0.50	≤0.08	0.25	0.20
GCr15	1.40 ~ 1.65	≤0.10	0.30	0.25
GCr15SiMn	1.40 ~ 1.65	≤0.10	0.30	0.25
GCr15SiMo	1.40 ~ 1.70	0.30 ~ 0.40	0.30	0.25
GCr18Mo	1.65 ~ 1.95	0.15 ~ 0.25	0.25	0.25

Because the Chromium natural resource is not rich in China, China developed the rolling bearing steel without Chromium partially substituting for Chromium rolling bearing steel partially. The rolling bearing steel without Chromium and their chemical compositions are listed in Table 1 – 90.

**Table 1 – 90 High-Carbon-without-Chromium bearing steel
chemical composition and their applictions**

grade	chemical composition (mass ratio, %)									usages
	C	Mn	Si	P	S	Cr	Ni	Mo	V	
GSiMnV	0.95 ~ 1.10	1.10 ~ 1.30	0.55 ~ 0.80	≤0.027	≤0.020	≤0.30	≤0.30	—	0.20 ~ 0.30	Substituting for GCr15
GSiMnMoV	0.95 ~ 1.10	0.75 ~ 1.05	0.40 ~ 0.65	≤0.027	≤0.020	≤0.30	≤0.30	0.20 ~ 0.40	0.20 ~ 0.30	Substituting for GCr15 SiMn
GMnMoV	0.95 ~ 1.07	1.10 ~ 1.40	0.15 ~ 0.40	≤0.027	≤0.020	≤0.30	≤0.30	0.40 ~ 0.60	0.15 ~ 0.25	Substituting for GCr15 SiMn

The High-Carbon-without Chromium bearing steel has good through-hardening, physical, forgeability characteristics and the fatigue life is as near the same as or little higher than the High-Carbon-Chromium Bearing Steel. But comparing with the High-Carbon-Chromium Bearing Steel, it is easy for decarburizing and has poor rustproof and corrosion resistance.

1. 5. 3 Carburizing Rolling Bearing Steel

For the rolling bearings that are subjected to impulse load, or the bearings that have particularly of large diameter and thin section thickness or the bearings of which working surface is required to have high wear resistance and high fatigue strength but its core part should have high strength and toughness, these bearing components should be made by carburizing bearing steel. The commonly used carburizing steel is Chromium-Nickel alloy and Manganese-Chromium alloy that their mass ratio of carbon is 0. 15%. The bearings made of carburizing bearing steel and through surface hardening treatment have almost the same characteristics as the bearings made of High-Carbon-Chromium Bearing Steel. So there is no need to differentiate them in bearing life calculation, which has been accepted by the ISO.

The commonly used applications of bearings made of carburizing bearing steel are listed in Table 1 – 91.

**Table 1 – 91 The commonly used applications of bearings
made of carburizing bearing steel**

Bearing type	select material	ring wall thickness/mm	roller diameter /mm	ball diameter /mm
common type bearing	G20CrNi3Mo	≤15	≤20	≤25
large type bearing	G20Cr2Ni4	≤70	≤100	≤130
extra-large type bearing	G15CrNi3Mo	≤110	≤140	≤170

1. Carburizing bearing steel and its chemical composition

Carburizing bearing steel and its chemical composition are listed in Table 1 – 92.

Table 1 – 92 Carburizing bearing steels and its chemical composition (GB/T3203)

grade	chemical composition (mass ratio, %)				
	C	Si	Mn	P ≤	S ≤
G20CrMo	0. 17 ~ 0. 23	0. 20 ~ 0. 35	0. 65 ~ 0. 95	0. 030	0. 030
G20CrNiMo	0. 17 ~ 0. 23	0. 15 ~ 0. 40	0. 60 ~ 0. 90	0. 030	0. 030
G20CrNi2Mo	0. 17 ~ 0. 23	0. 15 ~ 0. 40	0. 40 ~ 0. 70	0. 030	0. 030
G20Cr2Ni4	0. 17 ~ 0. 23	0. 15 ~ 0. 40	0. 30 ~ 0. 60	0. 030	0. 030
G10CrNi3Mo	0. 08 ~ 0. 13	0. 15 ~ 0. 40	0. 40 ~ 0. 70	0. 030	0. 030
G15CrNi3Mo	0. 13 ~ 0. 18	0. 15 ~ 0. 40	0. 40 ~ 0. 70	0. 030	0. 030
G20CrMn2Mo	0. 17 ~ 0. 23	0. 15 ~ 0. 40	1. 30 ~ 1. 60	0. 030	0. 030
G20CrMo	0. 35 ~ 0. 65	0. 08 ~ 0. 15	—	0. 25	
G20CrNiMo	0. 35 ~ 0. 65	0. 15 ~ 0. 30	0. 40 ~ 0. 70	0. 25	
G20CrNi2Mo	0. 35 ~ 0. 65	0. 20 ~ 0. 30	1. 60 ~ 2. 00	0. 25	
G20Cr2Ni4	1. 25 ~ 1. 75	—	3. 25 ~ 3. 75	0. 25	
G10CrNi3Mo	1. 00 ~ 1. 40	0. 08 ~ 0. 15	3. 00 ~ 3. 50	0. 25	
G15CrNiMo	1. 00 ~ 1. 40	0. 08 ~ 0. 15	3. 00 ~ 3. 50	0. 25	
G20CrMn2Mo	1. 70 ~ 2. 00	0. 20 ~ 0. 30	≤ 0. 30	0. 25	

2. The technical requirements for carburizing treatment

1) The requirements for carburizing layer depth are shown in Table 1 – 93.

Table 1 – 93 The requirements of carburizing layer depth for bearing components

Ring wall thickness	Carburizing layer depth	Roller diameter	Carburizing layer depth	Ball diameter	Carburizing layer depth
Medium, small bearing components					
6 ~ 7	1. 3 ~ 1. 6	7 ~ 10	1. 4 ~ 1. 6	≤ 4	0. 38 ~ 0. 65
8 ~ 10	1. 8 ~ 2. 2	11 ~ 14	1. 7 ~ 1. 9	5 ~ 8	0. 8 ~ 1. 3
11 ~ 14	2. 3 ~ 2. 7	15 ~ 19	2. 0 ~ 2. 2	9, 2	1. 4 ~ 1. 7
15 ~ 19	2. 8 ~ 3. 2			13 ~ 18	1. 8 ~ 2. 0
20 ~ 25	3. 3 ~ 3. 7	20 ~ 25	2. 3 ~ 2. 5	19 ~ 25	2. 1 ~ 2. 3

(continued)

Ring wall thickness	Carburizing layer depth	Roller diameter	Carburizing layer depth	Ball diameter	Carburizing layer depth
Extra-large bearing components					
< 700	≥4. 2	< 50	≥3. 5		
700 ~ 1000	≥4. 7	50 ~ 80	≥4. 0		
> 1000	≥5. 0	> 80	≥4. 5		

2) The surface hardness is 62 ~ 66 HRC after carburizing and hardening, and 60 ~ 64 HRC after annealing. The hardness of the core part should be greater than 25 HRC. No soft point and uneven hardness should be allowed on the surface.

3) A carburizing layer consists of immersed grains or fine, needle-like martensite and evenly distributed carbonization, and a little residual austenite, No coarse carbonization net and clearly visible carbonization needle are allowed. Fracture is not allowed after hardening and annealing.

4) After hardening and annealing, the fracture opening of carburizing layer of bearing components should be the opening of the grey, porcelain style, fine grains; the central fracture opening should be fiber style, and no coarse grain fracture is allowed.

5) The carbon density in the carburization layer should be controlled with in 0. 8% ~ 1. 0% . The carbon density gradient of the transitive layer should be smooth.

1. 5. 4 Medium Carbon Alloy Steel Used for Rolling Bearing Elements

Bearings working in impact condition can be made of medium carbon alloy steels in addition to carburization steels. The applications of the commonly used medium-carbon alloy steels are listed in Table 1 – 94.

Table 1 –94 The applications of medium-carbon alloy steels in making bearing components

Grade	Main application
50CrNiA	making spiral roller bearing components working in high impulse load condition
GCr10	making spiral roller bearing components working in high impulse load condition
65Mn	making spiral roller bearing outer ring with slot
55SiMoV	making rolling elements and rings in roller bit of petroleum and mine drilling machine
G8Cr15	making common bearing components or the bearing components under low impulse load

The medium-carbon alloy steel grade and composition for rolling bearings.

The medium-carbon alloy steel grade and composition for rolling bearings are shown in Table 1 –95.

Table 1 –95 The medium-carbon alloy steel grade and composition for rolling bearings

Grade	Composition (ratio, %)							
	C	Mn	Si	P	S	Cr	Ni	Mo
GCr10	0. 65 ~ 0. 75	≤0. 50	0. 15 ~ 0. 35	≤0. 035	≤0. 035	0. 90 ~ 1. 05	≤0. 30	<0. 10
50CrNiA	0. 47 ~ 0. 54	0. 50 ~ 0. 80	0. 17 ~ 0. 37	≤0. 035	≤0. 035	0. 45 ~ 0. 75	1. 00 ~ 1. 40	
65Mn	0. 62 ~ 0. 70	0. 90 ~ 1. 20	0. 17 ~ 0. 37	≤0. 035	≤0. 035	≤0. 25	≤0. 25	
G8Cr15	0. 75 ~ 0. 85	0. 20 ~ 0. 40	0. 15 ~ 0. 35	≤0. 027	≤0. 020	1. 30 ~ 1. 65	≤0. 30	

1. 5. 5 Steel for Special Rolling Bearings under Special Hostile Operating Conditions

1. Corrosion-resistance rolling bearing materials

The rolling bearings working in corrosive medium mostly are made of stainless steel. Usually for the rolling bearings working in oxidation medium, such as seawater, freshwater, vapor and wet condition or working in certain density of acidity or alkaline medium, the bearing rings and rolling elements can be made of stainless steel (9Cr18, 9Cr18Mo), and the cages can be made of 0Cr19Ni9 or 1Cr18Ni9 austenite stainless steel. If the rolling bearings work in high temperature and corrosive medium, most of the bearing rings and rolling elements are made of high temperature stainless bearing steel (Cr14Mo4). For large diameter corrosion-resistance bearings, most are made of 1Cr13, 2Cr13 or 3Cr13 half-martensite or martensite stainless steel after taking surface nitrogen treatment, and the cages are made of stainless steel or Beryllium Bronze.

For the bearings working in intense corrosive medium, such as dense acid, caustic soda and melt metal, usually these bearings only can be made of ceramic materials.

Because of the low hardness of stainless steel, the load capacity of bearings made of stainless steel is lower than that of the general bearings. Besides, the bearings made of stainless steel can be corrosion-resistance only under the conditions that all the surface polishing is very smooth and there are no colliding damage during installation.

Usually the corrosion-resistance bearings are made of stainless steel. The application of stainless steel for making rolling bearings listed in Table 1 – 96.

Table 1 – 96 Rolling bearing using stainless steel

Grade	Application
9Cr18	1. for making bearing rings and rolling elements working in corrosive mediums such as seawater, fresh water, distilled water, Nitrate, sea climate, vapor and strong oxidation atmosphere without lubrication

(continued)

Grade	Application
9Cr18Mo	2. for making miniature rolling bearing rings and rolling elements the 9Cr18Mo has higher hardness and higher stability of resistance to annealing than those of the 9Cr18 and has high temperature dimension stability
6Cr14Mo	3. suitable for making rolling bearing rings and rolling elements under working conditions of high vacuum and working within $-253 \sim 350^{\circ}\text{C}$ range 4. while 6Cr14Mo comparing with 9Cr18 (Mo), the fatigue strengthen could enhance about 1.5 ~ 2 times. The seawater corrosion resistance of these two alloys is equivalent. But in diluted nitric acid, corrosion resistance of the 6Cr14Mo alloy is lower than that of the 9Cr18 (Mo).
1Cr13	1. for making the outer rings of through bore drawn-cup needle roller bearings. After nitrogen treatment, these bearings could be under 650°C high temperature and corrosion resistance.
2Cr13	2. for making spherical bearing inner rings. After nitrogen treatment, the bearings could be under 600°C high temperature and corrosion resistance.
3Cr13	3. for making spherical bearing inner rings; for making bearings that could be under 400°C high temperature and corrosion resistance.
4Cr13	4. for making corrosion-resisting needle rollers and rings; for making the bearings which could be under 400°C high temperature and corrosion resistance.
1Cr17Ni2	5. for making high speed corrosion-resisting bearing cage and balls.
0Cr18Ni9	1. for making corrosion resistance bearing cage, shield, rivet, ring and ball.
1Cr18Ni9	2. After nitrogen treatment, the bearings can be used for large bearings under high temperature and corrosion resistance.
00Cr19Ni11	
1Cr18Ni9Ti	

The 9Cr18 and 9Cr18Mo alloys are the most commonly used materials for making acid-resisting, corrosion resistance rolling bearing rings and

rolling elements. Because, for 9Cr18 and 9Cr18Mo, it is easy to produce the coarse co-grain carbides during melting process, and it is not easy to change them after the follow-up heat treatment, the bearing fatigue life can be influenced by some extent, and the making of low noise bearings could be difficult. In recent years, Europe and the United States have developed 07Cr-13Cr (X65Cr13), China has developed 6Cr14Mo and Japan KOYO company has developed KUJ440C and so on, by lowering the content of C and Cr in the steel and reducing the co-grain carbide content and dimension. The rolling bearings made of these alloys have the better characteristics of the contact fatigue performance, hardness, impact toughness and salt-water corrosion resistance than the rolling bearings made of 9Cr18Mo (440C). But the corrosion resistance in diluted acid is not as good as 9Cr18Mo (440C). Moreover, Japan NSK company introduced the outstanding stainless bearing steel ES1 in 2000, and at the same time, Germany developed Cromindur 30 steel. These alloys are new steel containing nitrogen content (ω_N 0.4% about), and of which the corrosion resistance or all mechanical performance are clearly better than 9Cr18Mo (440), 07Cr-13Cr, X65Cr13 and 6Cr14Mo.

The commonly used stainless steel grade and chemical composition for making rolling bearing components in China are listed in Table 1 – 97.

Table 1 – 97 The stainless steel chemical composition for making rolling bearings

material and grade		chemical composition (mass ratio,%)							
		C	Mn	Si	P	S	Cr	Ni	Mo
stainless bearing steel	9Cr18	0. 90 ~ 1. 00	≤0. 80	≤0. 80	≤0. 027	≤0. 020	17. 00 ~ 19. 00	—	—
	6Cr14Mo	0. 6 ~ 0. 7	≤0. 80	≤0. 80	≤0. 027	≤0. 020	12. 5 ~ 14. 5	—	0. 40 ~ 0. 75
	9Cr18Mo	0. 95 ~ 1. 10	≤0. 80	≤0. 80	≤0. 027	≤0. 020	16. 00 ~ 18. 00	—	0. 40 ~ 0. 70

(continued)

material and grade		chemical composition (mass ratio, %)							
		C	Mn	Si	P	S	Cr	Ni	Mo
martensite stainless steel	1Cr13	≤0. 15	≤1. 00	≤1. 00	≤0. 035	≤0. 030	11. 50 ~ 13. 50	≤0. 60	—
	2Cr13	0. 16 ~ 0. 25	≤1. 00	≤1. 00	≤0. 035	≤0. 030	12. 00 ~ 14. 00	≤0. 60	—
	3Cr13	0. 26 ~ 0. 40	≤1. 00	≤1. 00	≤0. 035	≤0. 030	12. 00 ~ 14. 00	≤0. 60	—
	4Cr13	0. 45	≤1. 00	≤1. 00	≤0. 035	≤0. 030	12. 50 ~ 14. 50	≤0. 60	—
	1Cr17Ni2	0. 10 ~ 0. 17	≤0. 80	≤0. 80	≤0. 035	≤0. 030	16. 00 ~ 18. 00	1. 50 ~ 2. 50	—
austenite stainless steel	0Cr13Ni9	≤0. 08	≤2. 00	≤1. 00	≤0. 035	≤0. 030	18. 00 ~ 20. 00	9. 00 ~ 13. 00	—
	1Cr18Ni9	≤0. 15	≤2. 00	2. 00 ~ 3. 00	≤0. 035	≤0. 030	17. 00 ~ 19. 00	8. 00 ~ 10. 00	—
	00Cr19Ni11	≤0. 03	≤2. 00	≤1. 00	≤0. 035	≤0. 030	18. 00 ~ 20. 00	8. 00 ~ 10. 50	—
	1Cr18Ni9Ti	≤0. 12	≤2. 00	≤1. 00	≤0. 035	≤0. 030	17. 00 ~ 19. 00	8. 00 ~ 11. 00	Ti ≈ 0. 80

2. High temperature rolling bearing materials

High temperature rolling bearings are traditionally made of W18Cr4V or GCr4Mo4V and through hardening heat-treatment. But in the 1990's, Europe and the United States have developed the RBD, CBS1000 and M50NiL high temperature cementite bearing steels. The M50NiL steel has been widely used, after cementite and hardening, the bearing components of which have higher surface hardness, and their core toughness would be 2.5 times comparing with that made of M50 steel. At present, M50NiL mainly has been used for making bearings for aerospace engine spindle that have many requirements such as working in high temperature, high speed, high stress, and high reliability.

In addition to keeping the hardness under high temperature, the materials for making high temperature bearing components, must have wear resistance, fatigue resistance, anti-oxidation, corrosion resistance and anti-impact, and have good dimension stability and better machinability. The most commonly used high temperature bearing steel and its applications are listed in Table 1 – 98.

Table 1 – 98 High temperature bearing steel and its applications

Material and grade		Application
Bearing steel	GCr15	for making rings and rolling elements of working temperature < 200℃
Moderate temperature bearing steel	GCrSiWV	for making rings and rolling elements of working temperature ≤ 250℃
High temperature bearing steel	GCr4Mo4V	for making rings and rolling elements of working temperature ≤ 315℃
High temperature stainless bearing steel	GCr14Mo4	for making rings and rolling elements of working temperature < 330℃, and working in corrosive medium
High speed steel	W6Mo5Cr4V2	for making rings and rolling elements of working temperature ≤ 430℃
	W9Cr4V2Mo	for making rings and rolling elements of working temperature ≤ 450℃
	W18Cr4V	for making rings and rolling elements of working temperature ≤ 550℃

(continued)

Material and grade		Application
High temperature cementite bearing steel	H10Cr4Mo4Ni4V RR6027	for making larger cementite rings and rolling elements of working temperature $\leq 500^{\circ}\text{C}$
Special alloys	G60 G50	for making bearing rings and rolling elements working in high temperature, high pressure, water corrosive and low load condition

Besides, the bearings working in high temperature and corrosion resistance condition can be made of stainless steel through nitrogen treatment. See the corrosion resistance bearing materials for reference.

The most commonly used high temperature bearing steel grades and their composition are listed in Table 1 – 99.

The bearings working at the temperature higher than 120°C are called high temperature bearings, which are mainly used in aero-jet-engines, atomic energy reactors, X radiolucent tubes, the equipments of making semiconductors and varied melt-plating.

The components of the high temperature bearings made of GCr15 and GCr15SiMn must be through special high temperature annealing and dimension stabilization treatment. But if the annealing temperature is enhanced, the material hardness would be decreased and the bearing load capacity be reduced. In Table 1 – 100, the material hardness changes under different annealing temperatures for GCr15 and GCr15SiMn alloy steels are listed.

Table 1-99 The most commonly used high temperature bearing steel grades and their composition

material and grade	chemical composition(mass ratio, %)										
	C	Mn	Si	P	S	Cr	Ni	Mo	W	V	others
GCr15	0.95 ~ 1.05	0.20 ~ 0.40	0.15 ~ 0.35	≤0.027	≤0.020	1.30 ~ 1.65	≤0.30	—	—	—	—
GCrSiWV	0.95 ~ 1.05	0.40 ~ 0.60	0.70 ~ 0.90	≤0.027	≤0.020	1.30 ~ 1.60	≤0.30	—	1.10 ~ 1.30	0.20 ~ 0.35	
GCr4Mo4V	0.75 ~ 0.85	≤0.35	≤0.35	≤0.020	≤0.030	3.75 ~ 4.25	≤0.20	4.00 ~ 4.50	—	0.90 ~ 1.10	
GCr14Mo4	0.95 ~ 1.10	≤0.80	≤0.80	≤0.027	≤0.020	13.0 ~ 16.0	≤0.30	3.80 ~ 4.30	≤0.25	≤0.20	
W6Mo5Cr4V2	0.80 ~ 0.90	≤0.40	≤0.40	≤0.030	≤0.030	3.80 ~ 4.40	—	4.50 ~ 5.50	5.50 ~ 6.75	1.75 ~ 2.20	
W9Cr4V2Mo	0.90 ~ 1.00	≤0.40	≤0.40	≤0.030	≤0.030	3.80 ~ 4.40	—	≤1.00	8.50 ~ 10.00	2.00 ~ 2.60	
W18Cr4V	0.70 ~ 0.78	≤0.40	≤0.45	≤0.035	≤0.030	3.80 ~ 4.50	—	≤0.60	17.5 ~ 18.5	1.00 ~ 1.20	
RR6027	0.17 ~ 0.22	0.20 ~ 0.40	≤0.35	≤0.020	≤0.015	2.75 ~ 3.25	0.50 ~ 0.90	—	9.50 ~ 10.50	0.35 ~ 0.50	
G50	0.10 ~ 0.17	≤0.80	≤0.50	≤0.020	≤0.020	22.0 ~ 24.0	27.0 ~ 29.0	4.50 ~ 5.00	—	0.40 ~ 0.60	Ti ≈ 1.0 Al ≈ 1.0

Table 1 – 100 The hardness of the high temperature annealed bearing components

Annealing temperature /°C	annealing suffix code	hardness (HRC)					
		GCr15			GCr15SiMn		
		ring	ball	roller	ring	ball	roller
200	T	60 ~ 63	62 ~ 66 ^①	61 ~ 65 ^①	59 ~ 62	60 ~ 66 ^①	60 ~ 64 ^①
225	T ₁	59 ~ 62	62 ~ 66 ^①	61 ~ 65 ^①	59 ~ 62	60 ~ 66 ^①	60 ~ 64 ^①
250	T ₂	58 ~ 62	58 ~ 62 ^②	58 ~ 62 ^③	57 ~ 61	57 ~ 61 ^④	57 ~ 61 ^⑤
300	T ₃	55 ~ 59	55 ~ 59	55 ~ 59	54 ~ 58	54 ~ 58	54 ~ 58
350	T ₄	≥52	≥52	≥52	≥52	≥52	≥52
400	T ₅	≥48	≥48	≥48	≥48	≥48	≥48

① Usually there is no need to carry on high temperature annealing treatment. If the customer asks for high temperature annealing treatment, the hardness is the same as that of the ring.

② Carry on high temperature annealing treatment while the diameter is larger than 25mm.

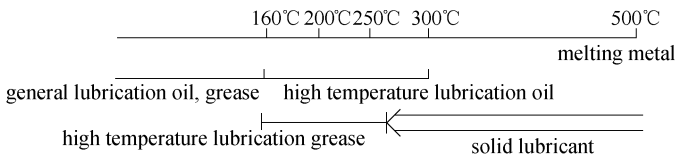
③ Carry on high temperature annealing treatment while the diameter is larger than 15mm.

④ Carry on high temperature annealing treatment while the diameter is larger than 50mm.

⑤ Carry on high temperature annealing treatment while the diameter is larger than 22mm.

The cage of the high temperature bearing usually is made of hard aluminum, brass or silicon-iron-bronze non-ferrous alloys.

Referring to Figure 1 – 36, the lubricant of the high temperature bearing could be selected.

**Figure 1 – 36 Rolling bearing lubricant selection according to working temperature**

3. Low temperature bearings

The rolling bearings of working temperature under -60°C are called low temperature bearings, which are typically used in all kinds of liquid-state gas pumps, such as liquefaction gas pump, liquid nitrogen (hydrogen, oxygen) pump, butane pump, liquid pump used in rocket missile, spaceship and so on.

The commonly used low temperature bearing is the single row deep groove ball bearing and cylindrical roller bearing.

Usually, the rings and rolling elements of low temperature bearings are made of stainless bearing steel (9Cr18 and 9Cr18Mo), and the cage is made of the same material or of polytetrafluoroethylene composite material.

There are two kinds of polytetrafluoroethylene composite materials:

- 1) 85% polytetrafluoroethylene plus 15% glass fiber.
- 2) 75% polytetrafluoroethylene plus 20% glass fiber and 5% molykote (MoS_2).

The failure mode of low temperature bearings mainly is the burn due to not good lubrication. The right lubricant must be selected carefully.

Low temperature bearings also could be made of Beryllium bronze and ceramic materials.

4. Sulphur — resisting bearing materials

The bearings working in the natural gas containing sulphuretted hydrogen (H_2S) must be made of the material having good sulphur-resisting characteristic, so that the bearing working in these medium could not have hydrogen fragility and electro-chemical corrosion.

The most commonly used sulphur-resisting material is the 00Cr40Ni55A13 alloy.

The 00Cr40Ni55A13 is a Nickel base alloy, whose composition is shown in Table 1 – 101.

Table 1 – 101 00Cr40Ni55A13 alloy composition (mass ratio, %)

C	Mn	Si	Ni	Cr	Al
≤0.03	≤0.80	≤0.80	52 ~ 58	37 ~ 42	2.5 ~ 3.4

The heat treatment parameters for 00Cr40Ni55A13 alloy are listed in Table 1 – 102.

Table 1 – 102 00Cr40Ni55A13 alloy heat treatment parameters

Material grade	Solid dissolve treatment				Time effectiveness			Hardness (HRC)
	temperature /℃	time /min	cooling	hardness (HRC)	temperature /℃	time/h	cooling	
00Cr40Ni-55A13	1100 ~ 1150	20 ~ 60	air cooling	< 23	680 ± 10	6.5	air cooling	51 ~ 55
				27 ~ 37	650 ~ 700	6.5	air cooling	> 50
				40 ~ 50	600 ~ 620	6.5	air cooling	> 50

If the bearings working in the natural gas containing sulphuretted hydrogen (H_2S) are made of general bearing steel, it is very easy to have hydrogen fragility and electro-chemical corrosion. These bearings must be made of the 00Cr40Ni55Al3 alloy material. This material is a kind of Ni-Cr alloy, whose hardness after heat treatment is 51 ~ 55 HRC and is lower than that of general bearings, and their load capacity is lower. Besides, the colliding damage and scratch on bearing component surface would greatly reduce the sulphur resistance, While using these bearings, more attention should be paid.

5. Anti-magnetic bearing materials

Anti-magnetic bearings must be made of the material whose magnetic conductivity is $\mu < 0.1$. The most commonly used anti-magnetic bearing materials and their applications are listed in Table 1 – 103.

Table 1 – 103 Anti-magnetic bearing materials and their applications

Material and grade	Application
QBz2 (铍青铜)	making bearing ring, rolling element and cage for anti-magnetic, low temperature, corrosion-resisting and low load bearings
25SiMnCr18Ni10WNbN NiCu28-2. 5-1. 5 (镍铜合金)	making anti-magnetic bearing inner ring, outer ring making anti-magnetic, corrosion-resisting bearing ball and cage
70Mn15Cr2Al3V2	making anti-magnetic bearing ring and rolling element
70Mn18Cr4V2WMoN	making anti-magnetic bearing ring and rolling element for petroleum roller bit
1Cr18Ni9Ti (不锈钢)	making anti-magnetic, corrosion-resisting bearing cage
G60 (防磁合金)	making bearing components working in high temperature, high vacuum, anti-magnetic and oxidation medium

The anti-magnetic materials and their composition are listed in Table 1 – 104.

Table 1-104 Anti-magnetic materials and their composition

Material grade	Chemical composition (mass ratio, %)										
	C	Mn	Si	P	S	Cr	Ni	Mo	W	V	Cu
QBe2	Be1.80 ~ 2.10	—	≤0.15	—	—	—	0.20 ~ 0.50	Pb ≤0.005	Al ≤0.15	Fe ≤0.15	remainder
NiCu28-2.5-1.5	≤0.3	1.20 ~ 1.80	≤0.50	—	≤0.020	—	Ni + Co	—	—	Fe2.0 ~ 3.0	27.0 ~ 29.0
G60	≤0.03	≤0.15	≤0.15	≤0.020	≤0.020	39.0 ~ 41.0	remainder	—	—	Ti0.90 ~ 1.15	Al 2.15 ~ 3.15
25SiMnCr18-Ni10WNbN	0.27 ~ 0.30	0.80 ~ 1.20	0.15 ~ 0.60	0.25 ~ 0.30	≤0.03	18.5 ~ 19.5	10.0 ~ 11.0	0.40 ~ 0.60	0.40 ~ 0.60	Nb0.30 ~ 0.40	N 0.15 ~ 0.25
70Mn15Cr2-Al3WMoV2	0.65 ~ 0.75	14.5 ~ 16.0	—	≤0.040	≤0.030	2.30 ~ 3.00	—	0.50 ~ 1.00	0.50 ~ 1.00	1.50 ~ 2.00	Al2.50 ~ 3.50
70Mn18Cr4V-2WMoN	0.65 ~ 0.75	17.0 ~ 19.0	0.50 ~ 1.00	—	—	3.50 ~ 4.50	—	0.40 ~ 0.60	0.40 ~ 0.60	Al 1.50 ~ 2.00	N ≤ 0.12
1Cr18Ni9(Ti)	≤0.12	≤2.00	≤1.00	0.035	0.030	17.0 ~ 19.0	8.00 ~ 11.00	—	—	—	(Ti ~ 0.8)

Anti-magnetic bearings are made of the non-magnetic material of very low magnetic conductivity rate. $\mu < 0.1$. The most commonly used anti-magnetic bearing material is Beryllium bronze (QBe2). It has very good mechanical, physical, chemical synthetic characteristics, would have high strength, hardness, elasticity, and wear resistance after hardening and quenching treatment, and has very good corrosion resistance under air, freshwater and seawater.

The load capacity of Beryllium Bronze (QBe2) bearing is very low, only as 1/4 as that of the general bearing made of general bearing steel. Recently there have been also developed some new anti-magnetic bearing materials, such as 25SiMnCr18Ni10WNbN, NiCu28-2.5-1.5, 70Mn15Cr-2Al3V2, 70Mn18Cr4V2WMoN, G60 and so on. The bearings made of these materials have advantages in anti-magnetic performance, and their load capacity and reliability have been enhanced in some extent comparing with

the QBe2 bearing.

It is valuable to mention that ceramic bearing has very good anti-magnetic performance.

6. High speed bearing materials

Usually when the bearing value $D_m n$ is over $1.0 * 10^6 \text{ mm} \cdot \text{r/min}$, this bearing is called the high speed bearing. Here, the D_m is the bearing mean diameter, and is the inner ring rotating speed. At present in practice, the $D_m n$ value for high speed bearings has arrived at $3.0 * 10^6 \text{ mm} \cdot \text{r/min}$. According to prediction, the $D_m n$ value for high speed bearings probably could be reaching $3.5 * 10^6 \text{ mm} \cdot \text{r/min}$.

The bearing types suitable for high speed rotation and the $D_m n$ value reached under different lubrication methods are listed in Table 1 – 105.

Table 1 – 105 High speed bearing type and $D_m n$ value

Bearing type	$D_m n$ value under different lubrication methods		
	spray oil	oil mist	oil air
deep groove ball bearing	$1.5 * 10^6$	$1.05 * 10^6$	$1.56 * 10^6$
angular contact ball bearing (contact angle 15°)	$1.5 * 10^6$	$1.10 * 10^6$	$1.56 * 10^6$
cylindrical roller bearing	$1.3 * 10^6$	$0.85 * 10^6$	$1.17 * 10^6$
soft roller bearing	$1.5 * 10^6$	—	—
precision ceramic ball bearing	$2.0 * 10^6$	—	$2.0 * 10^6$

While the bearing rotates in high speed, the centrifugal force has very large influence on the bearing life and strength. Therefore, the bearing high speed performance is always enhanced by reducing the rolling element mass. That is to select the special light or the extra light diameter series bearing, or to use the ceramic ball as rolling element. If there are requirements of high speed performance and high rigidity at the same time, usually the cylindrical roller bearing or the soft roller bearing is selected. The soft roller bearing is a new type cylindrical roller bearing developed recently, whose structure is shown in Figure 1 – 37. When mounting the soft roller bearing, three hollow rollers distributed equally are installed, whose diameters are larger than that of the solid rollers. The soft rollers are easily deformed under load, which could always keep contact with the inner ring and the

outer ring raceways. The contact force could let the cage have enough rotating dragging force, guaranteeing the stability of the cage rotating speed, and the evenly distributed bearing load.

The high speed bearing cage is the machined solid cage mostly made of the phenolic plastics of lamination cloth, polyimide or bronze. In order to reduce friction heat, silver-plating on the cage land surface can be used.

The high speed bearing accuracy should select the P4 and the P2 grade in order to perform the elastohydrodynamic lubrication (EHL). For large and medium type bearing of heavy load and high speed, the spray oil lubrication should be used; for small and miniature type bearing of light load and high speed, the oil mist lubrication or oil air lubrication should be used. Within the three lubrications, the cost of the spray oil is the highest, the cost of the oil mist lubrication is the lowest, and the oil air lubrication is a recently developed lubrication method. The basic difference of the oil air lubrication from the oil mist lubrication is that it could control the oil supply in the most proper amount, and the oil flows along the oil tube wall realizing continuous, small amount supply. While selecting lubrication oil, the bearing rotating speed should be considered. If the lubrication oil viscosity is too low, the oil film could not be fully formed. If the viscosity is too high, the viscous resistance would develop the heat, increasing the power loss. High speed bearing usually uses the low viscosity oil for lubrication. In Table 1 – 106, the most commonly used lubrication oi is listed.

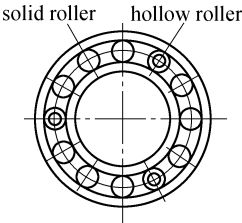


Figure 1 – 37 Soft roller bearing

Table 1 – 106 Commonly used high speed bearing lurication oil

Working temperature/℃	0 ~ 50	50 ~ 80	80 ~ 110
ISO viscosity grade	ISO VG 10	ISO VG 32	ISO VG 68
	ISO VG 15	ISO VG 46	ISO VG 100
	ISO VG 22	ISO VG 68	

The high speed bearing life would be lowered very quickly with the increase of the $D_m n$ value (sometimes also use the value dn , where d is the bearing inner ring diameter, mm, and n is the bearing rotating speed, r/min), The failure mode mainly is fatigue. The second failure is friction heat, causing the burning of the cage landing surface, the cage breaking and so on. Figure 1 – 38 shows the relationship between the dn value and the high speed bearing life.

7. Ceramic bearing materials

Ceramic bearings could be used in all of the hostile working conditions, such as high speed, high temperature, low temperature, intense corrosion, intense magnetic field, vacuum, high pressure and so on. For properly selecting and using ceramic bearing, the main characteristics of ceramic bearings are shown as follows:

High load capacity-The hardness of ceramic materials is 2 times of that of bearing steel, and the elastic modulus is 1.5 times of that of bearing steel. Therefore, the load capacity of a ceramic bearing is much higher than that of a bearing steel bearing.

High heat resistance The ceramic material strength has very small reduction under high temperature. Even under 800℃ , It also has very high strength and mechanical performance. At the same time, the coefficient of heat expansion of ceramic material only is 22% of that of bearing steel, so that it has very good dimension stability.

High rotating speed limit — The rotating speed limit of the ceramic ball bearing is over 1.3 times of that of the same dimension bearing made of bearing steel.

Small temperature rise due to friction heat — The ceramic bearing temperature rise under 1800r/min rotating speed is only about 60% of that

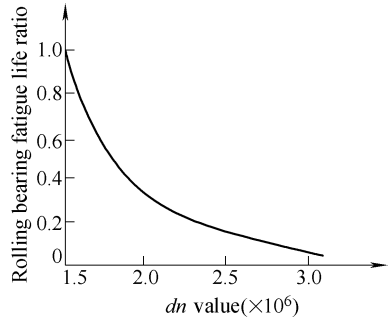


Figure 1 – 38 The relationship between the high speed bearing life and the dn value.

of the bearing made of bearing steel under the same rotating condition.

Small friction loss — For the same dimension bearing, the ceramic material mass is less than the steel (about 40% of the bearing steel). While rotating, the centrifugal force and the gyroscope moment applied to its rolling elements are smaller. Therefore the friction loss is clearly smaller than the bearing steel.

High durability — The ceramic material hardness is high, and the wear resistance is extremely good. The ceramic ball bearing life is two times of that of the general bearing steel bearing.

Good corrosion resistance — The ceramic material can resist all kinds of inorganic acid, organic acid, salt, alkali and melting metal corrosion such as hydrochloric acid (HCL), sulphuric acid (H_2SO_4), nitric acid (HNO_3), caustic soda (NaOH) and so on, However, its corrosion resistance is lower in hydrogen fluoride (HF) and melting iron, in which condition more attention should be paid.

Good insulation performance — The electrical resistance of ceramic material is $10^6 \Omega \cdot m$. Almost it is an insulating material, and its magnet conductivity of which is smaller than 0.1 H, which suites for making non-magnetic component.

Self-lubrication — The ceramic bearing could realize self-lubrication.

There are many kinds of ceramic materials, and the most commonly used materials in engineering are listed in Table 1 – 107. The most commonly used bearing material is the Nitrogen silicon.

Table 1 – 107 Engineering ceramic material names and chemical formula

Name	nitrogen-silicon	silicon carbide	titanium carbide	zirconium oxide	aluminum oxide
Chemical symbol	Si_3N_4	SiC	TiC	ZrO_2	Al_2O_3

8. Plastic bearing materials

Plastic material has very good performance in self-lubrication, wear resistance, low friction coefficient, and special interlock resistance. Even in bad lubrication condition, it still can normally work, Therefore, in many occasions, the plastic could substitute for metal for making bearings.

The commonly used engineering plastics for making bearing components

are listed in Table 1 – 108.

Table 1 – 108 Bearing using engineering plastics

Plastic name	Code	Name (English)
polyfomaldehyde	POM	polytormaldehyde (polyoxymethylene)
nylon 1010	PA	polyamide (nylon 1010)
nylon 6	PA	polyamide (nylon 6)
nylon 66	PA	Polyamide (nylon 66)
glass fiber intense nylon	GRPA	glass reinforced polyamide
extra high molecular polyethylene	UHMWPE	ultra high-molecular weight polyethylene
polymer	CP	chlorinated polyether (penton)
polymer	PC	polycarbonate
polymer	PPO	poly (phenylene oxide)
polymer	PSU	polysultone
polytetrafloroethylene	PTFE	polytetrafluoroethylene
polymer	PI	polyimide
phenolic lamination plastics	PF	phenol-formaldehyde resin

1.6 Commonly Used Rolling Bearing Technical Terms (vocabulary)

Along with the international commonuse of rolling bearings, the rolling bearing vocabulary has been in the international common use. The International Standard ISO 5593: 1997 stipulated the rolling bearing terms and their definitions. The China National Standard GB/T6390 — 2002 is the Chinese version.

The rolling bearing vocabulary can be divided into seven parts:

- 1) Vocabulary related to rolling bearings.
- 2) Vocabulary related to rolling bearing parts.
- 3) Vocabulary related to rolling bearing arrangements and subunits.
- 4) Vocabulary related to dimensions.
- 5) Vocabulary related to tolerance.

6) Vocabulary related to torque, load and life.

7) Vocabulary related to miscellaneous.

1. 6. 1 Vocabulary Terms Related to Rolling Bearings

1. Rolling bearings — General

(1) Rolling bearing

A bearing operating in rolling (rather than sliding) motion between the parts supporting load and moving in relation to each other, which comprises raceway members and rolling elements with or without spacing and/or guiding.

Note: It may be designed to support radial, axial or the combined radial and axial load.

(2) Single-row (rolling) bearing

A rolling bearing with one row of rolling elements.

(3) Double-row (rolling) bearing

A rolling bearing with two rows of rolling elements.

(4) Multi-row (rolling) bearing

A rolling bearing with more than two rows of rolling elements, supporting load in the same direction.

Note: It is preferable to specify the number of rows and type of bearing, for example; “four row (radial) cylindrical roller bearing”.

(5) Full complement (rolling) bearing

A rolling bearing without a cage, in which the sum of the clearances between the rolling elements in each row is less than the diameter of the rolling elements and is small enough to give satisfactory function of the bearing.

(6) Angular contact (rolling) bearing

A rolling bearing with a nominal contact angle greater than 0° but less than 90° .

(7) Rigid (rolling) bearing

A rolling bearing which resists misalignment between the axes of the raceways.

(8) Self-aligning (rolling) bearing

A rolling bearing which can accommodate angular misalignment and angular motion between the axes of its raceways due to one raceway being spherical.

(9) External-aligning (rolling) bearing

A rolling bearing which can accommodate angular misalignment between its axis and the axis of its housing by means of a spherical surface on ring or washer surface, which matches with a complementary seat surface in an aligning housing ring, in an aligning seat washer or in the housing.

(10) Separable (rolling) bearing

A rolling bearing with separable subunits.

(11) Non-separable (rolling) bearing

A rolling bearing from which after manufacture assembly the bearing ring can not be freely separated.

(12) Split (rolling) bearing

A rolling bearing with both rings and the cage, if used, are divided into two semicircular pieces to facilitate mounting.

Note: For a bearing with parts divided in a different manner, for example a bearing with a two-piece ring, there is no established short term.

(13) Metric (rolling) bearing

A rolling bearing originally designed with boundary dimensions and tolerances primarily in metric units.

(14) Metric series (rolling) bearing

A rolling bearing which conforms to a metric series of an ISO dimension.

(15) Inch (rolling) bearing

A rolling bearing originally designed with boundary dimensions and tolerances in inches.

(16) Inch series (rolling) bearing

A rolling bearing which conforms to an inch series of a dimension.

(17) Open (rolling) bearing

A rolling bearing with neither seals nor shields.

(18) Sealed (rolling) bearing

A rolling bearing which is fitted with a seal on one or both sides.

(19) Shielded (rolling) bearing

A rolling bearing which is fitted with a shield on one or both sides.

(20) Capped (rolling) bearing

A rolling bearing which is fitted with one or two seals, with one or two

shields or with one seal and one shield.

(21) Pre-lubricated (rolling) bearing

A rolling bearing which has been filled with lubricant by the manufacturer.

(22) Airframe (rolling) bearing

A rolling bearing which, by reason of design or function, is intended for use in the general structure of an aircraft, including its control systems.

(23) Instrument precision (rolling) bearing

A rolling bearing which, by reason of design or function, is intended for use in instruments.

(24) Railway axle-box (rolling) bearing

A rolling bearing which by reason of design or function is intended for use in railway axle-boxes.

Note: The most common type is a radial roller bearing.

(25) Matched (rolling) bearing

One of the rolling bearings in a matched pair or a matched stack.

2. Radial bearings

(1) Radial (rolling) bearing

A rolling bearing designed to support primarily radial load, having a nominal contact angle between 0° and 45° inclusive.

Note: Its principal parts are inner ring, outer ring and rolling elements with or without a cage.

(2) Radial contact (rolling) bearing

A radial rolling bearing with nominal contact angle of 0°

(3) Angular contact radial (rolling) bearing

A radial rolling bearing with a nominal contact angle greater than 0° up to 45° .

(4) Insert (rolling) bearing

A radial rolling bearing with a spherical outside surface and an extended inner ring with a locking device.

Note: It is primarily intended for use in a simple type of housing.

(5) Tapered bore (rolling) bearing

A radial rolling bearing with an inner ring with a tapered bore.

(6) Flanged (rolling) bearing

A radial rolling bearing with an external radial flange on one of its

rings, usually the outer ring or cup.

(7) Track roller (rolling bearing)

A radial rolling bearing with a heavy section outer ring, intended for use as a roller to roll on a track, for example a cam track.

(8) Yoke-type track roller (rolling bearing)

A track roller rolling bearing intended for mounting in a yoke.

(9) Stud-type track roller (rolling bearing)

A track roller rolling bearing in which the inner member is extended on one side in the form of a shaft for cantilever mounting of the bearing.

(10) Universal matching (rolling) bearing

A radial rolling bearing which, when used together with one or more similar bearing (s), selected randomly, yields predetermined characteristics in a paired or stack mounting.

3. Thrust bearings

(1) Thrust (rolling) bearing

A rolling bearing designed to support primarily axial load, having a nominal contact angle greater than 45° up to and including 90° .

Note: Its principal parts are shaft washer, housing washer and rolling elements with or without a cage.

(2) Axial contact (rolling) bearing

A thrust rolling bearing with a nominal contact angle of 90° .

(3) Angular contact thrust (rolling) bearing

A thrust rolling bearing with a nominal contact angle greater than 45° smaller than 90° .

(4) Single direction thrust (rolling) bearing

A thrust rolling bearing intended to support axial load in one direction only.

(5) Double-direction thrust (rolling) bearing

A thrust rolling bearing intended to support axial load in both directions.

(6) Double-row double-direction thrust (rolling) bearing

A double-direction thrust rolling bearing having two rows of rolling elements, each supporting axial load in one direction only.

4. Linear bearings

(1) Linear (motion) (rolling) bearing

A rolling bearing designed for relative linear motion between its

raceways in the direction of rolling.

(2) Re-circulating linear ball (roller) bearing

A linear motion rolling bearing is by means of the recirculation of the balls (rollers).

5. Ball bearings

(1) Ball bearing

A rolling bearing with balls as rolling elements.

(2) Radial ball bearing

A radial rolling bearing with balls as rolling elements.

(3) Groove ball bearing

A radial ball bearing the raceways of which are grooves generally with a cross-section of an arc of a circle with a radius slightly larger than the ball diameter.

(4) Deep groove ball bearing

A radial ball bearing in which each ring has uninterrupted raceway grooves with a cross-section: matching about one third of the circumference.

(5) Filling slot (ball) bearing

A groove ball bearing having a filling slot in one shoulder of each ring permits the insertion of a larger number of balls than in a deep groove ball bearing.

(6) Counter-bored ball bearing

A groove ball bearing with one outer ring shoulder completely or partly removed.

(7) Magneto (ball) bearing

A radial contact groove ball bearing with one outer ring shoulder completely removed, making this ring separable.

(8) Three-point-contact (ball) bearing

Single-row radial ball bearing in which, when under purely radial load, each loaded ball makes contact with one of the raceways at two points and with the other raceway at one point.

Note: Under pure axial load on the bearing, each ball makes contact with each raceway at only one point.

(9) Four-point-contact (ball) bearing

A single-row angular contact ball bearing in which, when under purely

radial load, each loaded ball makes contact with each of the two raceways at two points.

Note: Under pure axial load on the bearing, each ball makes contact with each raceway at only one point.

(10) Thrust ball bearing

A thrust rolling bearing with balls as rolling elements.

(11) Single-row double-direction thrust ball bearing

A four-point-contact ball bearing having contact angles greater than 45° .

(12) Double-row single-direction thrust ball bearing

A single-direction thrust rolling bearing having two concentric rows of balls both supporting load in the same direction.

6. Roller bearings

(1) Roller bearing

A roller bearing with rollers as rolling elements.

(2) Radial roller bearing

A radial rolling bearing with rollers as rolling elements.

(3) (radial) Cylindrical roller bearing

A radial rolling bearing with cylindrical rollers as rolling elements.

(4) (radial) Tapered roller bearing

A radial rolling bearing with tapered rollers as rolling elements.

(5) (radial) Needle roller bearing

A radial rolling bearing with needle rollers as rolling elements.

(6) Drawn-cup needle roller bearing

A radial needle roller bearing with a thin pressed outer ring (drawn cup), which may have one closed end or both ends open.

Note: The bearing is usually employed without an inner ring.

(7) (radial) Convex roller bearing

A radial rolling bearing with convex rollers as rolling elements.

(8) (radial) Concave roller bearing

A radial rolling bearing with concave rollers as rolling elements.

(9) (radial) Spherical roller bearing

A self-aligning, radial rolling bearing with convex rollers or concave rollers as rolling elements.

Not: With convex rollers, the outer ring has a spherical raceway, with concave rollers, the inner ring has a spherical raceway.

(10) Crossed roller bearing

An angular contact rolling bearing with one row of rollers. each roller positioned crosswise in relation to the adjacent rollers, such that an axial load in one direction is supported by half of the roller set (every second roller) whilst an axial load in the opposite direction is carried by the other half of the roller set.

(11) Thrust roller bearing

A thrust rolling bearing with rollers as rolling elements.

(12) Thrust cylindrical roller bearing

A thrust rolling bearing with cylindrical rollers as rolling elements.

(13) Thrust tapered roller bearing

A thrust rolling bearing with tapered rollers as rolling elements.

(14) Thrust needle roller bearing

A thrust rolling bearing with needle rollers as rolling elements.

(15) Thrust spherical roller bearing

A self-aligning, thrust rolling bearing with convex rollers or concave rollers as rolling elements.

Note: With convex rollers the housing washer has a spherical raceway; with concave rollers the shaft washer has a spherical raceway.

1. 6. 2 Terms Related to Rolling Bearing Parts

1. Bearing parts-General

(1) (rolling) Bearing part

One of the individual parts comprising a rolling bearing but excluding all accessories.

(2) (rolling) Bearing ring

An annular part of a radial rolling bearing incorporating one or more raceways.

(3) (rolling) Bearing washer

An annular part of a thrust rolling bearing incorporating one or more raceways.

(4) Separable bearing ring (bearing washer)

A bearing ring (bearing washer) which may be independently and freely separated from a complete rolling bearing.

(5) Interchangeable bearing ring (bearing washer)

A Bearing ring which can be replaced by another ring (another washer) of a similar group without impairing the function of the bearing.

(6) Single-split bearing ring

A bearing ring which is split or cut across its raceway (s), in one place only, to facilitate manufacturing assembly.

(7) Double-split bearing ring

A bearing ring which is split or cut across its raceway (s), in two places, to facilitate manufacturing assembly and/or mounting.

Note: The two splits are generally diametrical opposite each other.

(8) Two-piece bearing ring

A bearing ring divided into two annular pieces in a plane perpendicular to its axis, each piece incorporating at least part of a raceway.

(9) Loose rib

A separable basically flat washer, the outer or inner part of which serves as an inner ring rib or outer ring rib for a cylindrical roller radial bearing.

(10) (separable) Thrust collar

A separable ring having an L-shaped section, the outer part of which serves as an inner ring rib for a cylindrical roller radial bearing.

(11) Guide ring

A separable ring in a roller bearing with two or more rows of rollers, separating the rows and guiding the rollers.

(12) Locating snap ring

A single-split ring of constant cross section using a snap ring groove to locate axially a rolling bearing in its housing or on its shaft.

(13) Retaining snap ring

A single-split ring of constant cross section using a snap ring groove to serve as a rib retaining the rollers or the cage (with rolling elements) in a rolling bearing.

(14) Ring spacer

An annular part used between two bearing rings or bearing washers or

the two parts of a two-piece bearing ring or two-piece bearing washer to maintain a specified axial distance between them

(15) (bearing) Seal

A circular closure comprising one or several parts affixed to one bearing ring or bearing washer and extending towards the other ring or washer, with which it makes contact or forms a narrow labyrinth shaped gap, for the purpose of preventing leakage of lubricant or ingress of foreign substances.

(16) (bearing) Shield

A circular closure, usually of pressed sheet metal affixed to one bearing ring or bearing washer and extending towards the other ring or washer, covering the inner space but not making contact with the other ring or washer.

(17) Flinger

A component is attached to an inner ring or shaft washer, which by centrifugal action, the protection of a rolling bearing against ingress of foreign substances.

(18) Rolling element

A ball or roller which rolls between raceways.

(19) (rolling bearing) Cage

A bearing part which partly surrounds all of or several of the rolling elements and moves with them.

Note: Its purpose is to space the rolling elements and generally also to guide and/or retain them in the bearing.

(20) (rolling element) Separator

A bearing part which is placed between the adjacent rolling elements and moves with them, the main purpose being to space the rolling elements.

2. Features of bearing parts-General

(1) Raceway

A surface of a load supporting part of a rolling bearing, suitably prepared as a rolling track for the rolling elements.

(2) Straight raceway

A generatrix of which, in a plane perpendicular to the direction of rolling, is a straight line.

(3) Crowned raceway

A basically cylindrical or conical raceway which has a continuous,

slightly convex curvature in a plane perpendicular to the direction of rolling to prevent stress concentration at the ends of contacts between the rollers and the raceway.

(4) Spherical raceway

A raceway having the form of part of the surface of a sphere.

(5) Raceway groove

A raceway of a ball bearing in the form of a groove generally with a cross-section of an arc of a circle with a radius slightly larger than half the ball diameter.

(6) (groove) Shoulder

A flank of a raceway groove.

(7) Rib

A narrow ridge parallel to the direction of rolling projecting from a raceway surface, generally for the purpose of supporting and/or guiding and/or retaining the rolling elements in a rolling bearing.

(8) Cage riding land

A cylindrical surface of a bearing ring or bearing washer intended to guide a cage radially.

(9) Filling slot

A slot in a rib or a shoulder of a bearing ring or bearing washer permits the insertion of rolling elements.

(10) Face of a ring (a washer)

A surface of a ring (a washer) perpendicular to the axis of the ring (the washer).

(11) Bearing bore

A bore of the inner ring or shaft washer of a rolling bearing.

(12) Cylindrical bore

A bearing bore or bore of a bearing part whose generatrix is a basically straight line parallel to the bearing axis or the axis of the bearing part.

(13) Tapered bore

A bearing bore or bore of a bearing part whose generatrix is a basically straight line intersecting with the bearing axis or the axis of the bearing part.

(14) Bearing outside surface

An outside surface of the outer ring or the housing washer of a rolling bearing.

(15) Ring (washer) chamfer

A surface of a bearing ring (bearing washer) joining the bore or outside surface with one of the faces of the ring (the washer).

(16) Grinding undercut

A groove or slot at the root of a rib or flange of a bearing ring or bearing washer to facilitate grinding.

(17) Sealing (contact) surface

A surface with which a seal is intended to make sliding contact.

(18) Seal (shield) groove

A groove intended for the retention of a bearing seal (bearing shield).

(19) Snap ring groove

A groove to accommodate a locating snap ring or a retaining snap ring.

(20) Lubrication groove

A groove in a bearing part for conveying lubricant.

(21) Lubrication hole

A hole in a bearing part for conveying lubricant to the rolling elements.

3. Bearing rings

(1) (bearing) Inner ring

A bearing ring incorporating the raceway (s) on the outside surface.

(2) (bearing) Outer ring

A bearing ring incorporating the raceway (s) on the inside surface.

(3) Bearing cone

Note: This term has been replaced by (bearing) inner ring.

(4) (bearing) Cup

Note: This term has been replaced by (bearing) outer ring.

(5) Double inner ring

A bearing inner ring having two raceways.

(6) Double outer ring

A bearing outer ring having two raceways.

(7) Extended inner ring

A bearing inner ring extended on one or both sides in order to improve

the guidance of a shaft in its bore and/or permit the fixing of a locking device and/or to provide additional space for sealing devices.

(8) Stepped inner ring

An inner ring of a groove ball bearing with one shoulder completely or partly removed.

(9) Counterbored outer ring

A bearing outer ring of a groove ball bearing with one shoulder completely or partly removed.

(10) (bearing) Drawn cup

A bearing outer ring commonly of a needle roller radial bearing made of sheet metal drawn to shape with one end closed (closed end drawn cup) or both ends open.

(11) Flanged outer ring

A bearing outer ring with a flange (outer ring flange).

(12) Aligning outer ring

A bearing outer ring having a spherical outside surface to accommodate permanent angular misalignment between its axis and the axis of its housing.

(13) Aligning housing ring

A ring used between an aligning outer ring and a housing bore, having a spherical inside surface matching the spherical outside surface of the outer ring.

(14) Spherical outside surface

An outside surface, for example of a bearing outer ring, having the form of part of the surface of a sphere.

(15) Back face (of a bearing ring)

A face of a bearing ring which is intended to support axial load.

(16) Front face (of a bearing ring)

A face of a bearing ring which is not intended to support axial load.

(17) Outer ring flange

A flange on the outside of a bearing outer ring for the purpose of axially locating the bearing in its housing and to support axial load.

(18) (outer ring) Flange back face

A side of an outer ring flange which is intended to support axial load.

(19) Inner ring back face rib

(tapered roller bearing) A rib on the back face of the raceway of an inner ring, intended to guide the rollers and support the thrust load on the roller large end face.

(20) Inner ring front face rib

(tapered roller bearing) A rib on the front face of the raceway of an inner ring, intended to retain the rollers and, in the case of a bearing with an outer ring front face rib, to support the thrust on the roller small end face.

(21) Outer ring front face rib

(tapered roller bearing) A rib on the front face of the raceway of an outer ring intended to guide the rollers and support the thrust on the roller large end face.

(22) Centre rib

A central integral rib of a bearing ring with two raceway.

(23) Inner ring back face (front face) chamfer

A surface joining the back face (front face) of a bearing inner ring with the bore of the ring.

(24) Outer ring back face (front face) chamfer

A surface joining the back face (front face) of a bearing outer ring with the outside surface of the ring.

4. Bearing washer

(1) Shaft washer

A bearing washer which is intended to be mounted on a shaft.

(2) Housing washer

A bearing washer which is intended to be mounted in a housing.

(3) Central washer

A bearing washer having a raceway on each face used between the two rows of rolling elements of a double-row, double direction thrust rolling bearing.

(4) Aligning housing washer

A housing washer having a spherical back face to accommodate permanent angular misalignment between its axis and the axis of its housing.

(5) Aligning seat washer

A washer used between an aligning housing washer and the thrust-supporting surface of a housing, one face of which has a concave spherical surface matching the spherical back face of the aligning housing washer.

(6) Spherical back face

A housing washer back face which, or a zone of which, is convex with a spherical form.

(7) Shaft washer (housing washer) back face

A face of a shaft washer (a housing washer) intended to support axial load, generally opposite a raceway face.

(8) Shaft washer back face chamfer

A surface joining the shaft washer back face with the bore of the washer.

(9) Housing washer back face chamfer

A surface joining the housing washer back face with the outside surface of the washer.

5. Rolling elements

(1) Ball

A spherical rolling element.

(2) Roller

A rolling element having an axis of symmetry and being circle in cross-section in any plane perpendicular to that axis.

(3) Ball (roller) complement

All of the balls (the rollers) in a particular rolling bearing.

(4) Ball (roller) set

Balls (rollers) set in one row of a rolling bearing.

(5) Cylindrical roller

A generatrix of the outside surface of which is a basically straight line parallel to the roller axis.

(6) Needle roller

A cylindrical roller of small diameter with a large ratio of length to diameter.

Notes 1: It is generally accepted that the length is between three and ten times the diameter, which does not usually exceed 5mm.

Notes 2: The ends of a needle roller may be one of several shapes.

(7) Tapered roller

A generatrix of the outside surface of which is a basically straight line intersecting the roller axis, and the general shape is that of a truncated cone.

(8) Convex roller

An outside surface of which has a convex curvature in a plane containing its axis.

(9) Concave roller

An outside surface of which has a concave curvature in a plane containing its axis.

(10) Convex symmetrical roller

An outside surface of which is symmetrical about the plane perpendicular to the roller axis through the middle of the roller.

(11) Convex asymmetrical roller

An outside surface of which is asymmetrical about the plane perpendicular to the roller axis through the middle of the roller.

(12) Crowned roller

Basically cylindrical roller or basically tapered roller an outside surface of which has a continuous, slightly convex curvature in a plane containing the roller axis to prevent stress concentration at the ends of the contacts between the roller and the raceways.

(13) Relieved end roller

A roller with a slight modification of diameter at the ends of its outside surface to prevent stress concentration at the ends of the contacts between the roller and the raceways.

(14) Spiral wound roller

A roller made of winding steel strip in helical form.

(15) Roller end face

A surface at the end of a roller, basically perpendicular to the roller axis.

(16) Roller large end face

A face at the large end of a tapered roller or a convex asymmetrical roller.

(17) Roller small end face

A face at the small end of a tapered roller or a convex asymmetrical

roller.

(18) Roller recess

An indentation, dimple or undercut on the center of a roller end face.

(19) Roller chamfer

A surface joining the outside surface of a roller with one of the roller end face

6. Cages

(1) Ribbon cage

A rolling bearing cage comprising one or two corrugated annular parts.

(2) Snap cage

A rolling bearing cage with cage prongs the form of which permits the assembling of the cage with rolling elements by elastic deformation.

(3) Window cage

One piece roiling bearing cage having cage pockets surrounding the rolling elements.

(4) Prong cage

A one-piece rolling bearing cage with cage prongs.

(5) Pin cage

A two-piece cage with cage pins joining the two cage parts together.

(6) Two-piece cage

A rolling bearing cage comprising two annular parts joined together generally by rivets, cage pins or cage stays.

(7) Double-split cage

A rolling bearing cage which is split at two places to facilitate assembly. The two splits are generally diametrically opposite each other.

(8) Cage pocket

An aperture or gap in a rolling bearing cage to accommodate one or more roiling elements.

(9) Cage bar

A portion of a rolling cage separating the adjacent cage pocket.

(10) Cage prong

A cantilever cage bar projecting from the annular body of a rolling

bearing cage or from one part of a two-piece cage.

(11) Cage pin

A cage stay which is basically cylindrical and may pass through an axial hole in a roller.

(12) Cage stay

One of several parts used to join the two annular parts of a two-piece together and to maintain them at a specified distance from each other.

(13) Land-riding cage

A rolling bearing cage radially guided (centred) by a land on a bearing ring or bearing washer (cage riding land).

1. 6. 3 Terms Related to Rolling Bearing Arrangements and Subunits

1. Bearing arrangements

(1) Paired mounting

An arrangement of two rolling bearings mounted side by side on the same shaft such that they operate as a unit. The arrangement may be mounted back to back, face to face or tandem.

(2) Stack mounting

An arrangement of three or more rolling bearings mounted side-by-side on the same shaft such that they operate as a unit.

(3) Back-to-back arrangement

An arrangement of two rolling bearings mounted side-by-side with the back faces of their outer rings adjacent.

(4) Face-to-face arrangement

An arrangement of two rolling bearings mounted side-by-side with the front faces of their outer ring adjacent.

(5) Tandem arrangement

An arrangement of two or more rolling bearings mounted side-by-side with the back face of the outer ring of one bearing adjacent to the front face of the outer ring of the next bearing.

(6) Matched pair

Two rolling bearings which have been selected or manufactured to have predetermined characteristics, usually preload or clearance, when mounted

together in a specified way.

(7) Matched stack

Three or more rolling bearings which have been selected or manufactured to have predetermined characteristics, usually preload or clearance, when mounted together in a specified way.

2. Subunit

(1) Subunit

A bearing ring or bearing washer, with or without rolling elements or with cage and rolling elements, which can be freely separated from the bearing, or rolling element and cage assembly which can be freely separated from the bearing.

(2) Interchangeable subunit

A subunit which can be replaced by another subunit or a similar group without impairing the function of the bearing.

(3) Inner ring, cage and ball (roller) assembly

A subunit consisting of an inner ring, balls (rollers) and cage.

(4) Inner subunit

(tapered roller bearing) A subunit consisting of an inner ring, tapered rollers and cage.

(5) Outer ring, cage and ball (roller) assembly

A subunit consisting of an outer ring, balls (rollers) and cage.

(6) Needle roller bearing without inner ring

A subunit consisting of an outer ring and needle roller of a full complement bearing or an outer ring with needle rollers and cage.

Note: As required, an additional description of the bearing can be included in the term, for example “needle roller bearing, full complement, drawn cup, without inner ring” or “needle roller bearing with cage, machined ring, without inner ring”

3. Rolling element and cage assemblies

(1) Rolling element and cage assembly

A subunit consisting of rolling elements and cage of a rolling bearing.

(2) Ball (roller) and cage assembly

A subunit consisting of rolling elements and cage of a ball (a roller) bearing.

(3) Radial (thrust) ball and cage assembly

A ball and cage assembly of a radial (a thrust) ball bearing.

(4) Radial (thrust) roller and cage assembly

A roller and cage assembly of a radial (a thrust) roller bearing.

Note: As required, a description of the type of roller can be added to the term, for example “thrust needle roller and cage assembly” or “radial cylindrical roller and cage assembly”.

1.6.4 Terms Related to Dimensions

1. Dimension plans and series

(1) Dimension plan

System or table covering the boundary dimension of rolling bearings.

(2) Bearing series

Group of rolling bearings of a specific type, with gradually increasing dimensions and in most cases with the same contact angle and a certain relationship between boundary dimensions.

(3) Dimension series

Combination of a width series or a height series with a diameter series and, for tapered roller bearings, an angle series.

(4) Diameter series

Progressive series of bearing outside diameters, one for each standard bearing bore diameter, often with a specific relationship between the two diameters.

Note: Part of the ISO dimension plan.

(5) Width series

Progressive series of bearing width, for each bearing bore diameter of each diameter series.

Note: Part of the ISO dimension plan for radial bearings.

(6) Height series

Progressive series of bearing height, for each bearing bore diameter of each diameter series.

Note: Part of the ISO dimension plan for thrust bearings.

(7) Angle series

Specified range of contact angles.

Note: Part of the ISO dimension plan for tapered roller bearings.

2. Axes, Planes and directions

(1) Bearing axis

Theoretical axis of rotation of a rolling bearing.

Note: For a radial bearing this is the inner ring axis and for a thrust bearing the shaft washer axis.

(2) Inner ring (shaft washer) axis

Axis of the cylinder or cone inscribed in the basically cylindrical or tapered bore of an inner ring (a shaft washer).

(3) Outer ring (housing washer) axis

Axis of the cylinder circumscribed around the outside surface of an outer ring (housing washer), if this surface is basically cylindrical; the line perpendicular to the reference face of an outer ring, through the center of the sphere circumscribed around the outside surface of the ring, if this surface is basically spherical.

(4) (bearing) Cone (cup) axis

Replaced by inner (outer) ring axis of the tapered roller bearing.

(5) Radial plane

Plane perpendicular to an axis.

Note: For a bearing ring or a bearing washer it is generally acceptable to consider a radial plane as being parallel with the plane tangential to the reference face of the ring or the back face of the washer.

(6) Radial direction

Direction through an axis and in a radial plane.

(7) Axial plane

Plane containing an axis.

(8) Axial direction

Direction parallel with an axis.

Note: For a bearing ring or a bearing washer it is generally acceptable to consider an axial direction being perpendicular to the plane tangential to the reference face of the ring or the back face of the washer.

(9) Radial (axial) distance

Distance measured in a radial (an axial) direction.

(10) Contact angle (nominal contact angle)

Angle between a plane perpendicular to a bearing axis (a radial plane)

and the line of action (the nominal line of action) of the force transmitted by a bearing ring or washer to a rolling element.

(11) Load center

Point at which the resultants of the forces transmitted by a bearing ring or washer to the rolling element in one row intersect the bearing axis.

Note: This definition applies only if the contact angle is less than 90° and equal for all the rolling elements.

(12) Nominal contact point

Point on a raceway surface where a rolling element is intended to make contact when the bearing parts are in nominal relative positions.

(13) Reference face of a ring (a washer)

The face of a ring (a washer) is designated as the reference face by the manufacturer of the bearing and may be the datum for measurements.

3. Boundary dimensions

(1) (bearing) Boundary dimension

One of the dimensions (bore diameter, outside diameter, width or (height) and chamfer, etc.) defines the boundaries of a bearing.

(2) (bearing) Bore diameter

An inner ring bore diameter of a radial bearing or the shaft washer bore diameter of a thrust bearing.

(3) (bearing) Outside diameter

An outer ring outside diameter of a radial bearing or the housing washer outside diameter of a thrust bearing.

(4) (bearing) Width

An axial distance between the two ring faces designated to bound the width of a radial bearing.

Note: For a single row roller bearing, this is the axial distance between the back face of the outer ring and that of the inner ring.

(5) (bearing) Height

An axial distance between the two washer back faces designated to bound the height of a thrust bearing.

(6) Chamfer dimension

An extension in radial or axial direction of the surface of a ring (a

washer) chamfer.

(7) Radial chamfer dimension

A distance between the imaginary sharp corner of a ring or washer and the intersection of a chamfer surface and the face of the ring or washer.

(8) Axial chamfer dimension

A distance between the imaginary sharp corner of a ring or washer and the intersection of a chamfer surface and the bore or outside surface of the ring or washer.

(9) Flange width

An axial distance between the face of a flange.

(10) Flange height

A radial dimension of a flange.

Note: For an outer ring flange this is the radial distance between its outside surface and the outside surface of the outer ring.

(11) Snap ring groove diameter

A diameter of the cylindrical surface of a snap ring groove.

(12) Snap ring groove width

An axial distance between the faces of a snap ring groove.

(13) Snap ring groove depth

A radial distance between the cylindrical surface of a snap ring groove and the cylindrical surface in which the groove is machined.

(14) Radius of aligning surface

A radius of curvature of the spherical surface of an aligning housing washer, aligning seat washer, aligning outer ring or aligning housing ring.

(15) Center height of aligning surface

An axial distance between the center of curvature of the spherical back face of an aligning housing washer and the opposite shaft washer back face of a thrust bearing.

4. Dimensions of subunits and parts

(1) Raceway contact diameter

A diameter of the circle through the nominal contact points on a raceway.

(2) Middle of raceway

A point or line on a raceway surface, halfway between the two edges of

the raceway.

(3) Outer ring small inside diameter

(tapered roller bearing) A diameter of the imaginary circle of intersection of the back face of an outer ring and the cone inscribed in the outer ring, tangential to its raceway at the nominal contact points.

(4) Outer ring raceway angle

(tapered roller bearing) An angle included between two lines tangential to the raceway of an outer ring at the nominal contact points and in a plane containing the outer ring axis.

(5) Ring width

An axial distance between the two side faces of a rolling bearing ring.

(6) Washer height

An axial distance between the two outermost faces of a rolling bearing.

(7) Ball diameter

A distance between two parallel planes tangential to the surface of a ball.

(8) Roller Diameter

A distance between two tangents to the surface of a roller, parallel to each other and in a plane perpendicular to the roller axis (a radial plane).

Note: For the calculation of load rating the radial plane at the middle of the roller is used.

(9) Roller length

A distance between the two radial planes which just contain the extremities of the roller.

Note: For the calculation of load rating, however, the applicable “roller length” is the theoretical length of contact between a roller and that raceway where the contact is the shortest.

(10) Pitch diameter of ball set

Diameter of a circle containing the centers of the balls in one row in a bearing.

(11) Pitch diameter of roller set

Diameter of a circle intersecting the roller axes at the middle of the rollers in one row in a bearing.

(12) Ball set bore diameter (outside diameter)

Diameter of a cylinder inscribed inside (circumscribed around) one

row of balls in a bearing.

(13) Roller set bore diameter (outside diameter)

Diameter of a cylinder inscribed inside (circumscribed around) one row of rollers in a radial contact roller bearing.

(14) Ball complement bore diameter (outside diameter)

Diameter of a cylinder inscribed inside (circumscribed around) all of balls in a radial ball bearing.

(15) Roller complement bore diameter (outside diameter)

Diameter of a cylinder inscribed inside (circumscribed around) all of rollers in a radial contact roller bearing.

(16) Bore diameter of a (radial) ball (roller) and cage assembly

Theoretical ball (roller) complement bore diameter of a radial ball (roller) and cage assembly.

(17) Outside diameter of a (radial) ball (roller) and cage assembly

Theoretical ball (roller) complement outside diameter of a radial ball (roller) and cage assembly.

(18) Ball diameter of a thrust ball (roller) and cage assembly

Bore diameter of the cage of a thrust ball (roller) and cage assembly.

(20) Outside diameter of a thrust ball (roller) and cage assembly

Outside diameter of the cage of a thrust ball (roller) and cage assembly.

1. 6. 5 Terms Related to Tolerance

See Section 1. 4. 2.

1. 6. 6 Terms Related to Torque, Load and Life

1. Torque

(1) Starting torque

A torque required to start rotation of an bearing ring or washer with the outer ring or washer held stationary

(2) Running torque

A torque required to restrain motion of one bearing ring or washer while the outer ring or washer is rotated.

2. Actual load

(1) Radial load

A load acting in a direction perpendicular to the bearing axis.

(2) Axial load

A load acting in a direction parallel with the bearing axis.

(3) Centric axial load

An axial load of which the line coincides with the bearing axis.

(4) Static load

A load acting on a bearing when the speed of rotation of its rings or washers in relation to each other is zero (radial or thrust bearing) or when there is no motion in the direction of rolling between its raceway members (linear bearing).

(5) Dynamic load

A load acting on a bearing when its rings or washers rotate in relation to each other (radial or thrust bearing) or when there is motion in the direction of rolling between its raceway members (linear bearing).

(6) Stationary inner ring (shaft washer) load

A load which the line of action does not rotate in relation to the inner ring (shaft washer) of the bearing.

(7) Stationary outer ring (housing washer) load

A load which the line of action does not rotate in relation to the outer ring (housing washer) of the bearing.

(8) Rotating inner ring (shaft washer) load

A load which the line of action rotates in relation to the inner ring (shaft washer) of the bearing.

(9) Rotating outer ring (housing washer) load

A load which the line of action rotates in relation to the outer ring (housing washer) of the bearing.

(10) Oscillating load

A load of which the line of action, through an angle of less than 2π rad, in relation to one or both rings or washers of the bearing.

(11) Fluctuation load

A load of variable magnitude.

(12) Indeterminate direction load

A load of which the direction cannot be satisfactorily determined and

which therefore is considered as rotating or oscillating in relation to both rings or washers of the bearing.

(13) Preload

A force applied on a bearing before the “useful” load is applied.

Note: The force may be applied by axial adjustment against another bearing (external preload), or induced in a bearing by raceway and rolling element dimensions resulting in “negative clearance” (internal preload).

3. Equivalent load

(1) Equivalent load

General term for theoretical loads used in calculations and under which rolling bearings, in a specific respect, would react as they would under the actual load conditions.

(2) Static equivalent radial (axial) load

Static radial (static centric axial load) which would cause the same contact stress at the center of the most heavily loaded rolling element/raceway contact as that which occurs under the actual load conditions.

(3) Dynamic equivalent radial (axial) load

That constant stationary radial load (constant centric axial load) under the influence of which a rolling bearing would have the same life as it would attain under the actual load conditions.

(4) Mean effective load

Constant mean load under which a rolling bearing would have the same life as it would attain under a fluctuating load.

4. Load ratings

(1) Basic static radial (axial) load rating

Static radial load (static center axial load) corresponds to the following calculated contact stress at the center of the most heavily loaded rolling element/raceway contact.

4600MPa for self-aligning ball bearing;

4200MPa for all other radial ball bearing types and for thrust ball bearings;

4000MPa for all radial and thrust roller bearings.

Note: 1. In the case of a single row angular contact bearing, the radial load rating refers to

the radial component of that load which causes a purely radial displacement of the bearing ring in relation to each other.

2. For these contact stresses, a total permanent deformation of rolling element and raceway occurs which is approximately 0.0001 of the rolling element diameter.

(2) Basic dynamic radial (axial) load rating

That constant stationary radial load (constant center axial load) which a rolling bearing can theoretically endure for a basic rating life of one million revolutions.

Note: In the case of a single row angular contact bearing, the radial load rating refers to the radial component of that load which causes a purely radial displacement of the bearing ring in relation to each other.

5. Life

(1) Life

The number of revolutions (of an individual bearing) which one of the bearing rings or washer makes in relation to the other ring or washer before the first evidence of fatigue develops in the material of one of the rings or washers or one of the rolling elements.

Note: Life may also be expressed in number of hours of operation at a given constant speed of rotation.

(2) Reliability

(In the context of bearing life) For a group of apparently identical rolling bearings, operating under the same conditions, the percentage of the group that is expected to attain or exceed a specified life.

Note: The reliability of an individual rolling bearing is the probability that the bearing will attain or exceed a specified life.

(3) Median life

Life attained or exceeded by 50% of a group of apparently identical rolling bearings operating under the same condition.

(4) Rating life

Predicted value based on a basic dynamic axial load rating.

(5) Basic rating life

Rating life associated with 90% reliability.

(6) Adjusted rating life

Basic rating life adjusted for a reliability other than 90% , and/or non-

conventional operation conditions.

(7) Median rating life

Rating life associated with 50% reliability, i. e. the predicted median life based on a basic dynamic radial load rating or a basic dynamic axial load rating.

6. Calculation factors

(1) Radial (axial) load factor

Multiplication factor applied to the radial load (axial load) when calculating equivalent load

(2) Rotating factor

Multiplication factor which was sometimes applied (in addition to the radial load factor) to the radial load when calculating the dynamic equivalent load for a bearing with a rotating outer load.

Note: This factor is not used in International Standards.

(3) Life factor

Multiplication factor applied to the dynamic equivalent load in order to obtain the basic dynamic radial load rating or the basic dynamic axial load rating corresponding to a given rating life.

(4) Speed factor

Multiplication factor applied to the basic dynamic radial load rating or the basic dynamic axial load rating corresponding to a given rating life (expressed in hours of operation at a certain speed of rotation) in order to obtain the load rating corresponding to the same rating life at a different speed.

(5) Life adjustment factor

Multiplication factor applied to the basic rating life in order to obtain an adjusted rating life.

1.6.7 Other Terms

1. Housing

(1) (bearing) Housing

Part of a bearing mounting surrounds the bearing and has an inside surface matching the outside surface of the bearing outer ring or housing washer or of aligning housing ring or aligning seat washer.

(2) Plummer block

An assembly comprising a radial bearing and a bearing housing which

has a base plate with bolt holes for its mounting on a support surface normally parallel with the bearing axis.

(3) Plummer block housing

Bearing housing of plummer block (pillow block).

(4) Flanged housing

Bearing housing with a radial flange with bolt holes for its mounting on a support surface normally perpendicular to the bearing axis.

(5) Take-up housing

Bearing housing which has a facility for its adjustment relative to its support in a given direction normally perpendicular to the bearing axis.

2. Location and securing

(1) Bearing seating

That portion of a shaft on which is mounted or that portion of a bearing housing in which the bearing is mounted.

(2) Shaft (housing) shoulder

Portion of a shaft (housing) projects from the bearing seating surface and intended for axial location of the bearing.

(3) Adapter sleeve

An axial slotted sleeve with a cylindrical bore, a tapered outside surface and an external screw thread at its small end.

Note: It is used (with a locknut and a lockwasher) for the mounting of a bearing with tapered bore on a cylindrical surface of a shaft.

(4) Withdrawal sleeve

An axially slotted sleeve with a cylindrical bore, a tapered outside surface and an external screw thread at its large end.

Note: It is used for the mounting and dismounting (by means of a nut) of a bearing with tapered bore on a cylindrical surface of a shaft.

(5) Locknut

A nut (for axial location of rolling bearing) with a cylindrical outside surface with axial slots for locking the nut by one of the outer tabs of a lockwasher and for the application of a hook spanner.

(6) Lockwasher

A sheet steel washer (for locknut) with a plurality of outer tabs, one of

which is used for locking a locknut. One inner tab intended to enter the axial slot in an adapter sleeve or in a shaft.

(7) Eccentric locking

A steel ring having a recess on one side which is eccentric in relation to the bore and fits over an equally eccentric extension of the inner ring of an insert bearing.

Note: The collar is turned in relation to the inner ring until it locks and is then secured to the shaft by tightening a grub screw (seat screw, retaining screw)

(8) Concentric locking collar

A steel ring fitting over the extended inner ring of an insert bearing and having grub screws (set screw, retaining screws) which pass through holes in the inner ring to make contact with the shaft.

Chapter 2 Rolling Bearing Basic Load Rating and Life

Rolling bearings have been get comprehensive use in every machine. Notwithstanding the rolling bearings are the commonly used basic machinery elements that have been produced by standardization, serialization, and in a large amount, the correct selection, perfect usage, analyzing bearing running condition, analyzing the failure reasons, and ascertaining bearing load capacity and life calculation need many applied calculations.

Since the beginning of the rolling bearing industry, it has been put forward the question that how much load a rolling bearing could carry and how long a rolling bearing could be running under certain load? Early at the ending of the 19th century and at the beginning of the 20th century, some rolling bearing researchers had put the Hertz elastic contact theory into usage in rolling bearing calculation.

After several decades of probing research work and having been accumulating large amount test data, the Sweden G. Lundberg and A. Palmgren published their famous rolling bearing dynamic rating load theory. On the basis of their theory, the International Organization for Standardization, ISO, published the recommended standard ISO/R281-Rolling bearing dynamic load ratings and rating life in 1977, which later got an ascent as ISO281/I, and in 1990 revised as ISO281:1990. China adopted this standard that is equal to our national standard GB/T 6391 — 2003.

The rolling bearing basic dynamic load rating is a constant stationary load which a rolling bearing can theoretically endure for a basic rating life of one million revolutions. The rolling bearing basic static load rating is a constant stationary load which a rolling bearing can carry when its one ring is relatively static with another ring.

According to the theory of A. Palmgren, the ISO published the recommended standard ISO/R76-Rolling bearing static load ratings, which

later also got an ascent as ISO76.

This chapter mainly introduces the calculation method of the rolling bearing basic dynamic load ratings and rating life and the determination method of the rolling bearing static load ratings.

2.1 Fundamental Concepts

2.1.1 Contact Angle

The contact angle is a main design parameter of a rolling bearing. While analyzing the rolling bearing loads, deformations, motion relations and deciding rolling bearing load capacity, we need to decide the contact angle value at first.

The contact angle designates the angle between the rolling element force vector and the bearing radial plane^① at the center of the contact area of a rolling element and a raceway. The rolling element load acts on the center of the contact area, perpendicular to the contact area. Usually the contact angle designates the angle between the connecting line of the contact area center of a raceway with the rolling element center and the bearing radial plane.

Figure 2 – 1 shouis a sketch of bearing contact angle for several type rolling bearings.

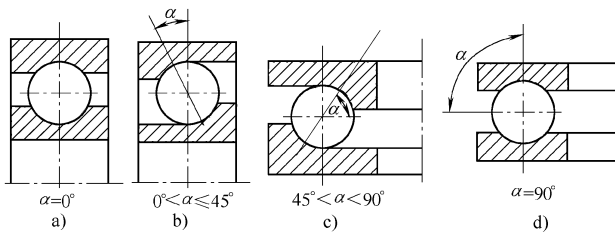


Figure 2 – 1 Bearing contact angle

- a) radial bearing b) angle contact bearing
c) thrust and radial bearing d) thrust bearing

① The plane through the rolling element center and perpendicular to the bearing axis is designated as bearing radial plane. The plane of including bearing axis is designated as axial plane.

$\alpha = 0^\circ$ designates radial bearing.

$0^\circ < \alpha \leq 45^\circ$ designates angular contact bearing or radial thrust bearing.

$45^\circ < \alpha < 90^\circ$ designates thrust radial bearing.

$\alpha = 90^\circ$ designates thrust bearing.

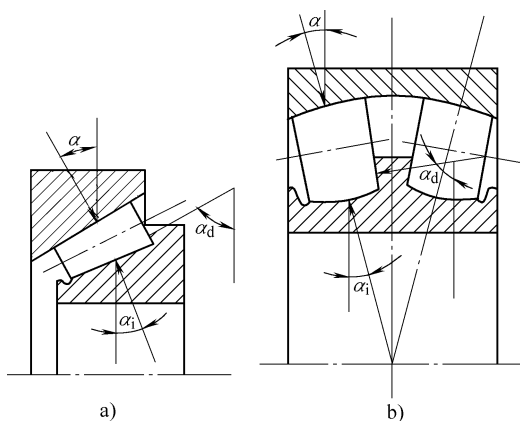


Figure 2-2 Tapered roller bearing and asymmetrical spherical roller bearing contact angle

a) tapered roller bearing b) asymmetrical spherical roller bearing

For tapered roller bearing and asymmetrical spherical roller bearing, the contact angle is different at the contact point of the rolling element contacting with different ring raceway. Generally, the contact angle of these type bearings designates the contact angle at the contact point of the rolling element contacting with the ring without rib, as shown in Figure 2-2.

In Figure 2-2:

α — bearing contact angle; because the ring without rib is the outer ring, it is the contact angle of the outer ring;

α_i — the contact angle of the inner ring raceway contact point contacting with rolling element;

α_d — the contact angle of the inner ring rib contacting with rolling element.

The contact angle of radial ball bearing and angular contact ball bearing

will increase when subjecting to axial load. In Section 2.5 of this chapter will discuss this in details.

2.1.2 Rolling Element Load

The subjecting load of a bearing is transferred from one ring to another ring. The value of every rolling element load is related to the bearing design parameter and load type (in this chapter omitting the rolling element inertia force).

1. Angular contact ball bearing

If omitting the inertia force, the contact angle at the inner ring contact point and at the outer ring contact point is the same, that is

$$\alpha_i = \alpha_e$$

From the equilibrium of the subjecting loads on a rolling element, the subjecting load Q_i from the inner ring and the subjecting load Q_e from the outer ring are the same but the directions are reverse, that is

$$Q_e = Q_i = Q$$

From Figure 2-3, we can get

$$Q = \frac{Q_r}{\cos \alpha} = \frac{Q_a}{\sin \alpha} \quad (2-1)$$

[**Example 2-1**] The angular contact ball bearing 7307C is subjected to pure axial load $F_a = 5\text{kN}$. Calculate the rolling element load on every ball? (assume under load, the contact angle is not changed. For 7307C bearing, its contact angle is $\alpha = 15^\circ$)

[**Resolve**] Because the bearing is subjected to pure axial load, the load on every ball is the same, and the ball numbers in 7307C bearing $z = 11$.

$$\text{So } Q_a = \frac{F_a}{z} = \frac{5\text{kN}}{11} = 0.455\text{kN}$$

From the equation (2-1) we can get:

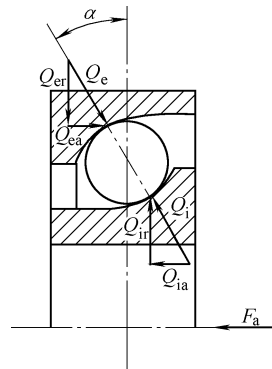


Figure 2-3 Angular contact ball bearing rolling element load

$$Q = \frac{Q_a}{\sin \alpha} = \frac{0.455 \text{ kN}}{0.2588} = 1.758 \text{ kN}$$

[**Example 2-2**] The angular contact ball bearing 7307B is subjected to pure axial load $F_a = 1 \text{ kN}$. Calculate the rolling element load on every ball? (assume under axial load, the contact angle is not changed. For 7307B bearing, its contact angle is $\alpha = 40^\circ$, the ball numbers in 7307B bearing $z = 11$).

[**Resolve**] $Q_a = \frac{F_a}{z} = \frac{1 \text{ kN}}{11} = 0.091 \text{ kN}$

From the equation (2-1) we can get:

$$Q = \frac{Q_a}{\sin \alpha} = \frac{0.091 \text{ kN}}{0.6428} = 0.1416 \text{ kN}$$

From Examples 1 and 2, we can see that, when the contact angle increases, the bearing axial load capacity will increase.

2. Tapered roller bearing and asymmetrical spherical roller bearing

Generally, the inner ring of the tapered roller bearing and the asymmetrical spherical roller bearing has a fixed rib, and the load of the rib acts on a roller, as shown in Figure 2-4.

The rolling element load from outer ring is:

$$Q_e = \frac{Q_{er}}{\cos \alpha_e} = \frac{Q_{ea}}{\sin \alpha_e} \quad (2-2)$$

The rolling element load from inner ring is divided as two parts:

1) At the inner ring contact point with a rolling element:

$$Q_i = \frac{Q_{ir}}{\cos \alpha_i} = \frac{Q_{ia}}{\sin \alpha_i} \quad (2-3)$$

2) At the inner rib contact point with the face of a rolling element:

$$Q_d = \frac{Q_{dr}}{\cos \alpha_d} = \frac{Q_{da}}{\sin \alpha_d} \quad (2-4)$$

Because all the forces acting on a rolling element should be equilibrium, we can get:

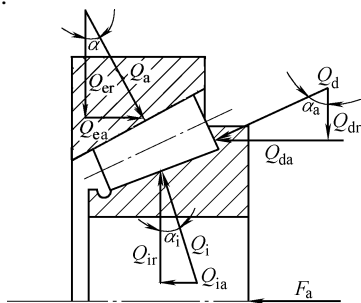


Figure 2-4 Tapered roller bearing rolling element load

$$\left. \begin{aligned} Q_{er} + Q_{dr} - Q_{ir} &= 0 \\ Q_{ea} - Q_{da} - Q_{ia} &= 0 \end{aligned} \right\} \quad (2-5)$$

Putting the equations (2-2) ~ (2-4) into (2-5), we can get:

$$Q_i = Q_e \left(\frac{\cos\alpha \sin\alpha_d + \sin\alpha \cos\alpha_d}{\cos\alpha_i \sin\alpha_d + \sin\alpha_i \cos\alpha_d} \right) \quad (2-6)$$

$$Q_d = Q_e \left(\frac{\sin\alpha \cos\alpha_i - \cos\alpha \sin\alpha_i}{\cos\alpha_d \sin\alpha_i + \sin\alpha_d \cos\alpha_i} \right) \quad (2-7)$$

Owing to generally in tapered roller bearing and asymmetrical spherical roller bearing, the outer ring is without rib. Therefore the bearing contact angle designates the contact angle of outer ring raceway with rolling element, that is

$$\alpha = \alpha_e$$

Knowing from bearing load distribution, we can calculate Q_e , and then again calculate Q_i and Q_d from Q_e .

If knowing that the bearing load is pure axial load, then the equation (2-6) and equation (2-7) can be rewritten as:

$$Q_i = Q_{ea} \frac{\cos\alpha_d + \cot\alpha \sin\alpha_d}{\sin(\alpha_i + \alpha_d)} \quad (2-8)$$

$$Q_d = Q_{ea} \frac{\cos\alpha_i - \cot\alpha \sin\alpha_i}{\sin(\alpha_i + \alpha_d)} \quad (2-9)$$

[**Example 2-3**] The 32306 tapered roller bearing subjects to pure axial load $F_a = 10\text{kN}$. Calculate the subjecting load on every roller. (The inner structure of the 32306 bearing are: $\alpha_e = 12^\circ$, $\alpha_i = 8^\circ$, $\alpha_d = 80^\circ$, $z = 12$).

[**Resolve**] Because the subjecting bearing load is a pure axial load, the subjecting axial load from the outer ring on every roller is:

$$Q_{ea} = \frac{F_a}{z} = \frac{10\text{kN}}{12} = 0.833\text{kN}$$

From the equation (2-2), we can get:

$$Q_e = \frac{Q_{ea}}{\sin\alpha_e} = \frac{0.833\text{kN}}{0.2079} = 4.007\text{kN}$$

From the equation (2-8), we can get:

$$Q_i = 0.833 \text{ kN} * \frac{\cos 80^\circ + \cot 12^\circ * \sin 80^\circ}{\sin(8^\circ + 80^\circ)} = 4.007 \text{ kN}$$

$$Q_d = 0.833 \text{ kN} * \frac{\cos 8^\circ - \cot 12^\circ * \sin 8^\circ}{\sin(8^\circ + 80^\circ)} = 0.336 \text{ kN}$$

2.1.3 Curvature Radius and Curvature

A segment of arc can be found at any point on a plane curve for substitution of the curve at the vicinity of this point. This radius of the arc is called the curvature radius. The reciprocal of this radius is called the curvature of this point.

That is:
$$\rho = \frac{1}{r} \quad (2-10)$$

The curvature has positive and negative. For the convex surface, its curvature is positive and for concave surface, its curvature is negative, as shown in Figure 2-5.



Figure 2-5 Expression method for curvature

2.1.4 Osculation and Ratio of Radius of Curvature

In bearing axial plane, the radius of curvature of raceway is little larger than the radius of curvature of rolling element. The difference of the curvature radius between raceway and rolling element can illustrate the osculation degree, expressed by S .

For ball bearing
$$S = \frac{D_w}{2r} \quad (2-11)$$

For roller bearing
$$S = \frac{R_w}{r} \quad (2-12)$$

In these equations:

D_w — rolling element diameter (mm);

r — raceway curvature radius (mm);

R_w — spherical curvature radius in axial plane (mm);

S — osculation.

In ball bearing design, using f designates the groove curvature radius size degree,

that is:
$$f = \frac{r}{D_w} \quad (2-13)$$

2.1.5 Principal Plane

For any curve surface body, the curvature in different plane passing through the contact point is different. The largest curvature plane and the smallest curvature plane are the principal planes, as shown in Figure 2-6. The curvatures of the two principle planes are expressed by the following symbols:

ρ_{11} — body I curvature in principle 1;

ρ_{12} — body I curvature in principle 2;

ρ_{21} — body II curvature in principle 1;

ρ_{22} — body II curvature in principle 2;

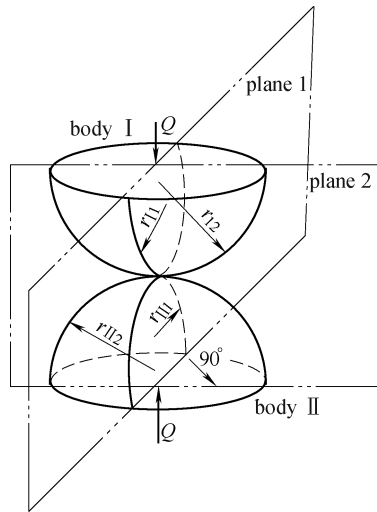
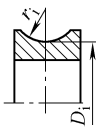
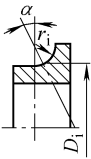
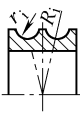
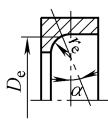
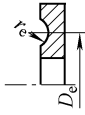


Figure 2-6 Principle plane

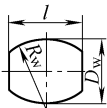
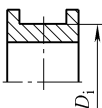
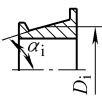
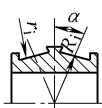
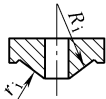
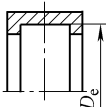
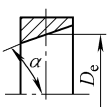
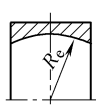
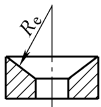
The first lower corner symbol expresses the body, and the second lower corner symbol expresses the principle plane. In rolling bearing, generally, the first principle plane designates the plane that passes through the contact area normal and is perpendicular with the bearing axial plane, which is the bearing radial plane. The second principle plane designates the bearing axial plane. The body I designates rolling element, the body II designates ring raceway.

The calculation equations of curvature for every type bearing are listed in Table 2-1.

Table 2 – 1 Calculation equations of curvature for every type bearing

Bearing type		cylindrical ball bearing	tapered roller bearing	aligning roller bearing	thrust aligning ball bearing
rolling element	sketch				
	ρ_{11}	$+\frac{2}{D_w}$			
	ρ_{12}	$+\frac{2}{D_w}$			
inner ring	sketch				
	ρ_{11}	$+\frac{2}{D_i}$	$+\frac{2\cos\alpha}{D_i}$	$+\frac{1}{R_i}$	0
	ρ_{12}	$-\frac{1}{r_i}$	$-\frac{1}{r_i}$	$-\frac{1}{r_i}$	$-\frac{1}{r_i}$
outer ring	sketch				
	ρ_{11}	$-\frac{2}{D_e}$	$-\frac{2\cos\alpha}{D_e}$	$-\frac{1}{R_e}$	0
	ρ_{12}	$-\frac{1}{r_e}$	$-\frac{1}{r_e}$	$-\frac{1}{r_e}$	$-\frac{1}{r_e}$

(continued)

Bearing type		cylindrical ball bearing	tapered roller bearing	aligning roller bearing	thrust aligning ball bearing
rolling element	sketch				
	ρ_{I1}	$+\frac{2}{D_w}$	$+\frac{2}{D_{wc}}$	$+\frac{2}{D_w}$	
	ρ_{I2}	0	0	$+\frac{1}{R_w}$	
inner ring	sketch				
	ρ_{II1}	$+\frac{2}{D_i}$	$+\frac{2\cos\alpha_i}{D_i}$	$+\frac{1}{R_i}$	$+\frac{1}{R_i}$
	ρ_{II2}	0	0	$-\frac{1}{r_i}$	$-\frac{1}{r_i}$
outer ring	sketch				
	ρ_{II1}	$-\frac{2}{D_e}$	$-\frac{2\cos\alpha}{D_e}$	$-\frac{1}{R_e}$	$-\frac{1}{R_e}$
	ρ_{II2}	0	0	$-\frac{1}{R_e}$	$-\frac{1}{R_e}$

2.1.6 Curvature Sum $\Sigma \rho$ and Function $F(\rho)$

The $\Sigma \rho$ is sum of the principle curvatures of two contact bodies.

$$\Sigma \rho = \rho_{I1} + \rho_{I2} + \rho_{II1} + \rho_{II2} \quad (2-14)$$

The symbols are shown in Figure 2-7.

In which $D_e = D_{pw} + D_w \cos \alpha \quad (2-15)$

$$D_i = D_{pw} - D_w \cos \alpha \quad (2-16)$$

If let

$$\gamma = \frac{D_w \cos \alpha}{D_{pw}} \quad (2-17)$$

Thus the curvature sum in ball bearing can be calculated from the following equation:

$$\Sigma \rho = \frac{1}{D_w} \left(4 \pm \frac{2\gamma}{1 \mp \gamma} - \frac{1}{f} \right) \quad (2-18)$$

In this equation, the upper symbol is used for the inner ring raceway contact point with rolling element, and the lower symbol is used for the outer ring raceway contact point with rolling element.

The contact surface shape of two contacting bodies is related to the function $F(\rho)$. The $F(\rho)$ value could be decided by the principle curvatures and the angle between the two principle planes of the two contacting bodies. For rolling bearing, this angle is zero,

So

$$F(\rho) = \frac{(\rho_{I1} - \rho_{I2}) + (\rho_{II1} - \rho_{II2})}{\Sigma \rho} \quad (2-19)$$

In rolling bearing, the $F(\rho)$ always is positive.

For ball bearing
$$F(\rho) = \frac{\frac{1}{f} \pm \frac{2\gamma}{1 \mp \gamma}}{4 \pm \frac{2\gamma}{1 \mp \gamma} - \frac{1}{f}} \quad (2-20)$$

In this equation, the upper symbol is used for the inner ring raceway contact point with rolling element, and the lower symbol is used for the outer ring raceway contact point with rolling element.

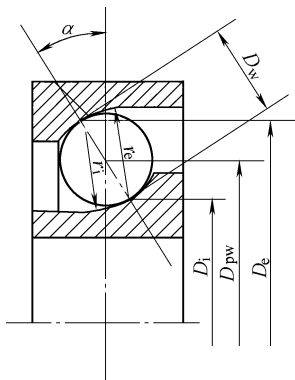


Figure 2-7 Ball Bearing

[**Example 2-4**] To calculate the $\Sigma \rho$ and $F(\rho)$ of the angular contact ball bearing 7208C. Having known: $D_w = 12.7\text{mm}$, $\alpha = 12^\circ$, $\gamma_i = \gamma_e = 6.54\text{mm}$, $D_e = 72.42\text{mm}$. In that, D_e is the diameter of the outer ring raceway contact point with rolling element; D_i is the diameter of the inner ring raceway contact point with rolling element; D_w is the ball diameter; D_{pw} is the pitch diameter of ball center.

[**Resolve**] From the equation (2-13), we can get:

$$f_i = f_e = \frac{6.54}{12.7} = 0.515$$

From the equation (2-15), we can get:

$$D_{pw} = D_e - D_w \cos \alpha = 60\text{mm}$$

From the equation (2-17), we can get:

$$\gamma = \frac{D_w \cos \alpha}{D_{pw}} = 0.207$$

At the inner ring contact point with ball, from the equations (2-18) and (2-20) we can get:

$$\begin{aligned} \Sigma \rho_i &= \frac{1}{D_w} \left(4 + \frac{2\gamma}{1-\gamma} - \frac{1}{f_i} \right) \\ &= \frac{1}{12.7} \left(4 + \frac{2 * 0.207}{1 - 0.207} - \frac{1}{0.515} \right) = 0.203 \\ \Sigma \rho_e &= \frac{1}{D_w} \left(4 - \frac{2\gamma}{1+\gamma} - \frac{1}{f_e} \right) \\ &= \frac{1}{12.7} \left(4 - \frac{2 * 0.207}{1 + 0.207} - \frac{1}{0.515} \right) = 0.135 \\ F(\rho_i) &= \frac{\frac{1}{0.515} + \frac{2 * 0.207}{1 - 0.207}}{4 + \frac{2 * 0.207}{1 - 0.207} - \frac{1}{0.515}} = 0.955 \\ F(\rho_e) &= 0.932 \end{aligned}$$

2.1.7 Contact Types

For two rotation bodies having different radii of curvature in a pair of principle planes passing through the contact between the bodies under the

condition of no applied load, basically there are two contact types:

Point contact — That is two surfaces touch each other at a single point. Such a condition is called point contact.

Line contact — That is two surfaces touch each other at a straight line or a curve line. Such a condition is called line contact.

1. Point contact

Obviously, after a load is applied to the contacting bodies, the point contact will expand to a contact surface, and the projection of this contact surface on the plane perpendicular to the contact area normal is an ellipse. The major axis is denoted by $2a$, the minor axis is denoted by $2b$, as shown in Figure 2-8.

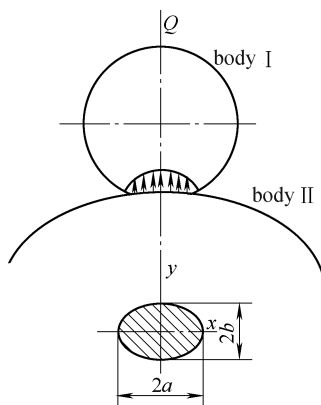


Figure 2-8 Contact ellipse

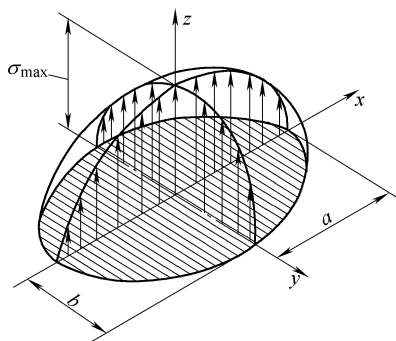


Figure 2-9 The stress distribution in contact area

In the contact area, the contact stress distribution is an ellipse. If putting the major axis as the coordinate x and putting the minor axis as the coordinate y , the coordinate z is perpendicular to the plane including the coordinates x and y , as shown in Figure 2-9. The contact stress at any point in the contact area is:

$$\sigma = \frac{3Q}{2\pi ab} \left[1 - \left(\frac{x}{a} \right)^2 - \left(\frac{y}{b} \right)^2 \right]^{1/2} \quad (2-21)$$

From Equation (2-21), the maximal contact stress at the contact area center is as follows: ($x=0$, $y=0$)

$$\sigma_{\max} = \frac{3Q}{2\pi ab} \quad (2-22)$$

2. Line contact

For ideal line contact condition, that is the bodies have equal length, after a load is applied to the contacting bodies, the contact line is expanded to a rectangular area, and the width of the contact area is denoted by $2b$, as shown in Figure 2-10.

The contact stress distribution in contact area is in accordance with half ellipsoid, as shown in Figure 2-11. The contact stress at any point of the contact area is

$$\sigma = \frac{2Q}{\pi lb} \left[1 - \left(\frac{y}{b} \right)^2 \right]^{1/2} \quad (2-23)$$

The contact stress on the center line is the maximum, and its value is:

$$\sigma_{\max} = \frac{2Q}{\pi lb} \quad (2-24)$$

The idea line contact exists only between two infinite long and parallel cylinders. On the contact condition of two equal length cylinders of finite length, the contact stress distribution along with these cylinders only can be approximately considered as even distribution. For roller bearings, the raceway width is larger than the roller length. In this

situation, the axial stress distribution along the roller is varied, since the material in the raceway is in tension at the roller ends because of compression of the raceway outside of the roller ends. The roller end compressive stress tends to be higher than that in the center of contact. This phenomenon is called as end edging stress, as shown in Figure 2-12.

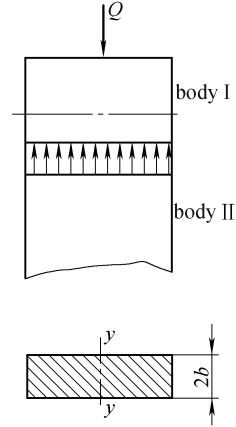


Figure 2-10 Line contact

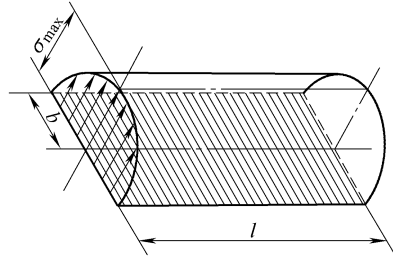


Figure 2-11 Contact stress distribution in the contact area

Because the bearing life is related to the $10/3^{\text{th}}$ power of the load in roller bearing, and the load is also related to the square of contact stress, the edging stress would influence on the bearing load capacity very much.

In order to enhance the bearing load capacity, the contact generatrix shape must be modified for reducing or eliminating the edging stress. Generally, the modifying method is to let the middle part of a raceway becoming a little crowning, let the generatrix becoming a convex curve, or modify the roller end part shape, as shown in Figure 2 – 13.

Experience has been shown that for roller bearings, after the contacting generatrix modifying, the bearing life can be enhanced and the load capacity can be increased.

For the roller bearing through modifying contacting generatrix, the edging stress only can be eliminated in a certain load range. One modifying amount just corresponds to an optimum load. If the applied bearing load is lower than the optimum load, there would be no edging stress. If the applied bearing load is larger than the optimum load, there would be edging stress again. Thus there are three kinds of contact conditions for roller bearing:

1) Point contact condition When there is no applied load, the roller contacts with raceway at a point. When there is applied load, but the major axis $2a$ of the contact ellipse is smaller than or equal to the effective contact length l_{we} , this condition is called as point contact. At this contact condition:

$$2a < l_{we}$$

2) Modifying line contact When the major axis $2a$ of the contact

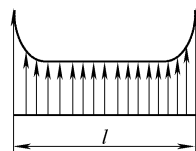
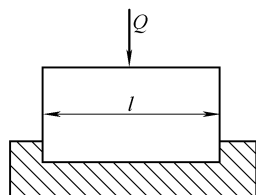


Figure 2 – 12 The edging stress

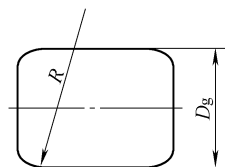


Figure 2 – 13 Modifying roller

ellipse is $l_{we} < 2a < 1.5 l_{we}$, this condition is called as modifying line contact.

3) Line contact When $2a > 1.5 l_{we}$, this contact condition is called as line contact. In this condition, the edging stress also exists.

In roller bearing design, in order to eliminate the edging stress enhancing the load capacity, it is the way to select the roller and raceway curvature radii according to the modifying contact condition.

2.1.8 Relative Elastic Approach Quantity δ of Two Bodies

Because there are deformations between two contacting bodies, the space positions of the two contacting bodies are in different places before and after contacting. The space position change of the two contacting bodies can be expressed by the relative elastic approach quantity δ of two remote fix points (apart from the contacting area some distance) on two contacting bodies in the load direction.

2.2 Contact Stress and Deformation in Rolling Bearings

2.2.1 General Introduction

In order to make decision of the bearing load capacity, it always needs to calculate the contact stress and deformation in rolling bearings. Besides, under the working condition of heavy load and exceeding heavy load in a short period, the contact stress and deformation in rolling bearings have to be checked.

Loads acting between two curved bodies develop only small areas of contact between the mating bodies. The contact area shape is related to the curvatures of the two contacting bodies and the contact area size is related to the acting loads. In rolling bearings, the contacts between the rolling elements and the raceways are the curve surface bodies. The classical solution for local stress and deformation of two elastic bodies apparently contacting at a single point was established by Hertz. So the Hertz elastic theory can apply to the calculation of the rolling bearing contact stress and deformation.

The hypotheses of Hertz elastic bodies contact theory are as follows:

- 1) The materials of the contacting bodies must be isotropic materials. That is the material strength is uniform and consistence.
- 2) The contacting surfaces of the contacting bodies should be absolute smooth. When the contacting bodies are in static position, there is only a normal force perpendicular to the contacting surfaces but without shearing force.
- 3) The contact area size is very small comparing with the surfaces of the contacting bodies.
- 4) The contacting deformation is within the material elastic limit range.

For rolling bearings, the aforementioned hypotheses are not fully conformed. For instance, the rolling bearing steel is not isotropic, but having non-metal impurities; strictly saying, the surfaces of the contacting bodies are not absolute smooth, but there exists plastic deformation even under not so much contact stress condition; in the high osculation ball bearing, the contact area is not very small comparing with the contacting surfaces.

However, the experiences have been verified that the Hertz theory can be used for computing the contact stress and the deformation in rolling bearings, and the results of the theoretical calculation are conformed with the experimental results.

2.2.2 Point Contact

The fundamental equations of the elastic deformation and contact stress derived from the Hertz theory are as follows:

$$a = \mu \sqrt[3]{\frac{3}{E} \left(1 - \frac{1}{m^2}\right) \frac{Q}{\sum \rho}}$$

$$b = \nu \sqrt[3]{\frac{3}{E} \left(1 - \frac{1}{m^2}\right) \frac{Q}{\sum \rho}}$$

$$\sigma_{\max} = \frac{3}{2} \frac{Q}{\pi ab}$$

$$\delta = 1.5 \frac{2K}{\pi} \frac{\left(1 - \frac{1}{m^2}\right)}{Ea} Q$$

In these equations:

a, b — contact ellipse semi-major and semi-minor axes (mm);

$\sigma_{\max}$ — maximum contact stress (N/mm²);

δ — elastic approach quantity (mm);

μ, ν — ellipse integral relating to the curvature function $F(\rho)$, listed in Table 2-2;

Q — normal load tightly pressing two contact bodies each other, in rolling bearings as rolling element load;

$\Sigma \rho$ — principal curvature sum on contact point.

Table 2-2 $F(\rho)$ relating with $\mu\nu$

$F(\rho)$	μ	ν	$\mu\nu$	$2K/\pi\mu$
0.9995	23.95	0.163	3.91	0.171
0.9990	18.53	0.185	3.43	0.207
0.9985	15.77	0.201	3.17	0.230
0.9980	14.25	0.212	3.02	0.249
0.9975	13.15	0.220	2.89	0.266
0.9970	12.26	0.228	2.80	0.279
0.9965	11.58	0.235	0.72	0.291
0.9960	11.02	0.241	2.65	0.302
0.9955	10.53	0.246	2.59	0.311
0.9950	10.15	0.251	2.54	0.320
0.9945	9.77	0.256	2.50	0.328
0.9940	9.46	0.260	2.46	0.336
0.9935	9.17	0.264	2.42	0.343
0.9930	8.92	0.268	2.39	0.350
0.9925	8.68	0.271	2.36	0.356

(continued)

$F(\rho)$	μ	ν	$\mu\nu$	$2K/\pi\mu$
0.9920	8.47	0.275	2.33	0.362
0.9915	8.27	0.278	2.30	0.368
0.9910	8.10	0.281	2.28	0.373
0.9905	7.93	0.284	2.25	0.379
0.9900	7.76	0.287	2.23	0.384
0.9895	7.62	0.289	2.21	0.388
0.9890	7.49	0.292	2.19	0.393
0.9885	7.37	0.294	2.17	0.398
0.9880	7.25	0.297	2.15	0.402
0.9875	7.13	0.299	2.13	0.407
0.9870	7.02	0.301	2.11	0.411
0.9865	6.93	0.303	2.10	0.416
0.9860	6.84	0.305	2.09	0.420
0.9855	6.74	0.307	2.07	0.423
0.9850	6.64	0.310	2.06	0.427
0.9845	6.55	0.312	2.04	0.430
0.9840	6.47	0.314	2.03	0.433
0.9835	6.40	0.316	2.02	0.437
0.9830	6.33	0.317	2.01	0.440
0.9825	6.26	0.319	2.00	0.444
0.9820	6.19	0.321	1.99	0.447
0.9815	6.12	0.323	1.98	0.450
0.9810	6.06	0.325	1.97	0.453
0.9805	6.00	0.327	1.96	0.456
0.9800	5.94	0.328	1.95	0.459
0.9795	5.89	0.330	1.94	0.462
0.9790	5.83	0.332	1.93	0.465
0.9785	5.78	0.333	1.92	0.468
0.9780	5.72	0.335	1.92	0.470
0.9775	5.67	0.336	1.91	0.473

(continued)

$F(\rho)$	μ	ν	$\mu\nu$	$2K/\pi\mu$
0.9770	5.63	0.338	1.90	0.476
0.9765	5.58	0.339	1.89	0.478
0.9760	5.53	0.340	1.88	0.481
0.9755	5.49	0.342	1.88	0.483
0.9750	5.44	0.343	1.87	0.486
0.9745	5.39	0.345	1.86	0.489
0.9740	5.35	0.346	1.85	0.491
0.9735	5.32	0.347	1.85	0.493
0.9730	5.28	0.349	1.84	0.495
0.9725	5.24	0.350	1.83	0.498
0.9720	5.20	0.351	1.83	0.500
0.9715	5.16	0.353	1.82	0.502
0.9710	5.13	0.354	1.81	0.505
0.9705	5.09	0.355	1.81	0.507
0.9700	5.05	0.357	1.80	0.509
0.969	4.98	0.359	1.79	0.513
0.968	4.92	0.361	1.78	0.518
0.967	4.86	0.363	1.77	0.522
0.966	4.81	0.365	1.76	0.526
0.965	4.76	0.367	1.75	0.530
0.964	4.70	0.369	1.74	0.533
0.963	4.65	0.371	1.73	0.536
0.962	4.61	0.374	1.72	0.540
0.961	4.56	0.376	1.71	0.543
0.960	4.51	0.378	1.70	0.546
0.959	4.47	0.380	1.70	0.550
0.958	4.42	0.382	1.69	0.553
0.957	4.38	0.384	1.68	0.556
0.956	4.34	0.386	1.67	0.559
0.955	4.30	0.388	1.67	0.562

(continued)

$F(\rho)$	μ	ν	$\mu\nu$	$2K/\pi\mu$
0.954	4.26	0.390	1.66	0.565
0.953	4.22	0.391	1.65	0.568
0.952	4.19	0.393	1.65	0.571
0.951	4.15	0.394	1.64	0.574
0.950	4.12	0.396	1.63	0.577
0.948	4.05	0.399	1.62	0.583
0.946	3.99	0.403	1.61	0.588
0.944	3.94	0.406	1.60	0.593
0.942	3.88	0.409	1.59	0.598
0.940	3.83	0.412	1.58	0.603
0.938	3.78	0.415	1.57	0.608
0.936	3.73	0.418	1.56	0.613
0.934	3.68	0.420	1.55	0.618
0.932	3.63	0.423	1.54	0.622
0.930	3.59	0.426	1.53	0.626
0.928	3.55	0.428	1.52	0.630
0.926	3.51	0.431	1.51	0.634
0.924	3.47	0.433	1.50	0.638
0.922	3.43	0.436	1.50	0.642
0.920	3.40	0.438	1.49	0.646
0.918	3.36	0.441	1.48	0.650
0.916	3.33	0.443	1.47	0.653
0.914	3.30	0.445	1.47	0.657
0.912	3.27	0.448	1.46	0.660
0.910	3.23	0.450	1.45	0.664
0.908	3.20	0.452	1.45	0.667
0.906	3.17	0.454	1.44	0.671
0.904	3.15	0.456	1.44	0.674
0.902	3.12	0.459	1.43	0.677
0.900	3.09	0.461	1.42	0.680

(continued)

$F(\rho)$	μ	ν	$\mu\nu$	$2K/\pi\mu$
0.895	3.03	0.466	1.41	0.688
0.890	2.97	0.471	1.40	0.695
0.885	2.92	0.476	1.39	0.702
0.880	2.86	0.481	1.38	0.709
0.875	2.82	0.485	1.37	0.715
0.870	2.77	0.490	1.36	0.721
0.865	2.72	0.494	1.35	0.727
0.860	2.68	0.498	1.34	0.733
0.855	2.64	0.502	1.33	0.739
0.850	2.60	0.507	1.32	0.745
0.84	2.53	0.515	1.30	0.755
0.83	2.46	0.523	1.29	0.765
0.82	2.40	0.530	1.27	0.774
0.81	2.35	0.537	1.26	0.783
0.80	2.30	0.544	1.25	0.792
0.75	2.07	0.577	1.20	0.829
0.70	1.91	0.607	1.16	0.859
0.65	1.77	0.637	1.13	0.884
0.60	1.66	0.664	1.10	0.904
0.55	1.57	0.690	1.08	0.922
0.50	1.48	0.718	1.06	0.938
0.45	1.41	0.745	1.05	0.951
0.40	1.35	0.771	1.04	0.962
0.35	1.29	0.796	1.03	0.971
0.30	1.24	0.824	1.02	0.979
0.25	1.19	0.850	1.01	0.986
0.20	1.15	0.879	1.01	0.991
0.15	1.11	0.908	1.01	0.994
0.10	1.07	0.938	1.00	0.997
0.05	1.03	0.969	1.00	0.999
0	1	1	1	1

In order to simplify the calculation, introducing the factors C_E , C_a , C_b , C_σ , C_δ .

(1) Material quality factor C_E

Taking
$$E_0 = \frac{E}{1 - \frac{1}{m^2}}; E_{0I} = \frac{E_I}{1 - \frac{1}{m_I^2}};$$

$$E_{0II} = \frac{E_{II}}{1 - \frac{1}{m_{II}^2}}; \frac{1}{E_0} = \frac{1}{2} \left(\frac{1}{E_{0I}} + \frac{1}{E_{0II}} \right)$$

We Can get
$$a = \mu \sqrt[3]{\frac{3Q}{2 \sum \rho} \left(\frac{1}{E_{0I}} + \frac{1}{E_{0II}} \right)}$$

$$b = \nu \sqrt[3]{\frac{3Q}{2 \sum \rho} \left(\frac{1}{E_{0I}} + \frac{1}{E_{0II}} \right)}$$

Taking
$$C_E = \sqrt[3]{\frac{E_0}{2} \left(\frac{1}{E_{0I}} + \frac{1}{E_{0II}} \right)} \quad (2-25)$$

We Can get
$$a = C_E \mu \sqrt[3]{\frac{3}{E_0} \frac{Q}{\sum \rho}}$$

$$b = C_E \nu \sqrt[3]{\frac{3}{E_0} \frac{Q}{\sum \rho}}$$

If the materials of the two contacting bodies are the same, that is, the E and the $1/m$ are the same value, then, $C_E = 1$. In Table 2-3 the values of varied materials are listed.

Table 2-3 C_E values

Contacting body material	$E/$ (N/mm ²)	m	C_E	C_E^2	$C_E^{3/2}$	$C_E^{2.7}$
steel	$2.079 * 10^5$	$\frac{10}{3}$	1	1	1	1
steel	$2.079 * 10^5$	$\frac{10}{3}$				
steel	$2.079 * 10^5$	$\frac{10}{3}$	1.11	1.24	1.20	1.40
cast iron	$1.128 * 10^5$	4				
cast iron	$1.128 * 10^5$	4	1.24	1.53	1.38	1.78
cast iron	$1.128 * 10^5$	4				

(continued)

Contacting body material	$E/ \text{ (N/mm}^2 \text{)}$	m	C_E	C_E^2	$C_E^{3/2}$	$C_E^{2.7}$
steel	$2.079 * 10^5$	$\frac{10}{3}$	1.18	1.39	1.28	1.56
brass	$0.892 * 10^5$	3				
brass	$0.892 * 10^5$	3	1.32	1.73	1.51	2.10
steel	$2.079 * 10^5$	$\frac{10}{3}$				
bronze	$1.079 * 10^5$	3	1.13	1.28	1.20	1.39
bronze	$1.079 * 10^5$	3				
bronze	$1.079 * 10^5$	3	1.23	1.52	1.37	1.77

(2) Ellipse major and minor axes factors C_a , C_b

Taking
$$C_a = \mu \sqrt[3]{\frac{3}{E_0}}; \quad C_b = \nu \sqrt[3]{\frac{3}{E_0}}$$

We can get
$$a = C_a C_E \sqrt[3]{\frac{Q}{\Sigma \rho}} \quad (2-26)$$

$$b = C_b C_E \sqrt[3]{\frac{Q}{\Sigma \rho}} \quad (2-27)$$

If the contacting bodies both are made by steel, $E = 2.079 * 10^5 \text{ N/mm}^2$, $m = \frac{10}{3}$, We can get $C_a = 0.0236\mu$; $C_b = 0.0236\nu$.

(3) Contact stress factor C_σ

$$\sigma_{\max} = \frac{3Q}{2\pi ab} = \frac{3}{2\pi C_E^2 C_a C_b} \frac{Q}{\sqrt[3]{\left(\frac{Q}{\Sigma \rho}\right)^2}}$$

Taking
$$C_\sigma = \frac{3}{2\pi C_a C_b} = \frac{3}{2\pi \mu \nu \sqrt[3]{\left(\frac{3}{E_0}\right)^2}}$$

We can get
$$\sigma_{\max} = C_\sigma C_E^{-2} \sqrt[3]{Q (\Sigma \rho)^2} \quad (2-28)$$

If the contacting bodies both are made by steel, $E = 2.079 * 10^5 \text{ N/mm}^2$, $m = \frac{10}{3}$, then $C_\sigma = 857.27 \frac{1}{\mu \nu}$.

(4) Elastic approach quantity factor C_{δ}

$$\delta = 1.5 \frac{2K}{\pi} \frac{1 - \frac{1}{m^2}}{Ea} Q = \frac{2K}{\pi} \frac{1.5Q}{E_0 C_a C_E \sqrt[3]{\frac{Q}{\Sigma\rho}}}$$

Taking
$$C_{\delta} = \frac{2K}{\pi\mu} \frac{1.5}{E_0 \sqrt[3]{\frac{3}{E_0}}}$$

We Can get
$$\delta = C_{\delta} C_E^{-1} \sqrt[3]{Q^2 \Sigma\rho} \quad (2-29)$$

If the contacting bodies both are made by steel, $E = 2.079 * 10^5 \text{ N/mm}^2$, $m = \frac{10}{3}$, then $C_{\delta} = 2.78 * 10^{-4} \frac{2K}{\pi\mu}$.

$$a = C_a \sqrt[3]{\frac{Q}{\Sigma\rho}} \quad (2-30)$$

$$b = C_b \sqrt[3]{\frac{Q}{\Sigma\rho}} \quad (2-31)$$

$$\sigma_{\max} = C_{\sigma} \sqrt[3]{Q (\Sigma\rho)^2} \quad (2-32)$$

$$\delta = C_{\delta} \sqrt[3]{Q^2 \Sigma\rho} \quad (2-33)$$

When the contact bodies both are made by the same material, $C_E = 1$.

Each factor is $C_a = 0.0236\mu$; $C_b = 0.0236\nu$; $C_{\sigma} = 857.27 \frac{1}{\mu\nu}$; $C_{\delta} = 2.78 * 10^{-4} \frac{2K}{\pi\mu}$.

The C_a , C_b , C_{σ} and C_{δ} are listed in Table 2-4. μ and ν is the ellipse integral related to the curvature function $F(\rho)$, which can be found from Table 2-2; So, C_a , C_b , C_{σ} , C_{δ} can be found from Table 2-4, and also can be calculated from μ , ν . Remember that the presupposition of using the factors in Table 2-4 is that the contacting bodies both are made by steel. That is: $E = 2.079 * 10^5 \text{ N/mm}^2$, $m = \frac{10}{3}$.

Table 2-4 Factors C_a , C_b , C_σ , C_δ values

$F(\rho)$	C_a	C_b	C_σ	C_δ
0.9995	0.5652	0.003847	219.60	0.00004753
0.9990	0.4373	0.004366	250.07	0.00005756
0.9985	0.3722	0.004744	270.45	0.00006388
0.9980	0.3363	0.005003	283.77	0.00006932
0.9975	0.3103	0.005192	296.32	0.00007390
0.9970	0.2893	0.005381	306.68	0.00007761
0.9965	0.2733	0.005546	315.02	0.00008088
0.9960	0.2601	0.005688	322.79	0.00008393
0.9955	0.2485	0.005806	330.94	0.00008655
0.9950	0.2395	0.005924	336.49	0.00008894
0.9945	0.2306	0.006042	342.75	0.00009134
0.9940	0.2233	0.006136	348.54	0.00009352
0.9935	0.2164	0.006230	354.11	0.00009548
0.9930	0.2105	0.006325	358.61	0.00009745
0.9925	0.2048	0.006396	364.44	0.00009897
0.9920	0.1999	0.00649	368.04	0.0001007
0.9915	0.1952	0.006561	372.88	0.0001025
0.9910	0.1912	0.006632	376.64	0.0001038
0.9905	0.1877	0.006702	380.65	0.0001055
0.9900	0.1831	0.006773	384.92	0.0001068
0.9895	0.1798	0.006820	389.28	0.0001079
0.9890	0.1768	0.006891	391.97	0.0001092
0.9885	0.1739	0.006938	395.64	0.0001107
0.9880	0.1711	0.007009	398.13	0.0001118
0.9875	0.1683	0.007056	402.12	0.0001131
0.9870	0.1657	0.007104	405.71	0.0001142
0.9865	0.1635	0.007151	408.26	0.0001158
0.9860	0.1614	0.007198	410.92	0.0001168
0.9855	0.1591	0.007245	414.30	0.0001177
0.9850	0.1567	0.007316	416.47	0.0001188

(continued)

$F(\rho)$	C_a	C_b	C_σ	C_δ
0.9845	0.1546	0.007363	419.49	0.001197
0.9840	0.1532	0.007410	421.97	0.0001206
0.9835	0.1510	0.007458	423.89	0.0001216
0.9830	0.1494	0.007481	427.22	0.0001223
0.9825	0.1477	0.007528	429.29	0.0001236
0.9820	0.1461	0.007576	431.44	0.0001243
0.9815	0.1444	0.007623	433.67	0.0001251
0.9810	0.1430	0.00767	435.27	0.0001260
0.9805	0.1416	0.007717	436.94	0.0001269
0.9800	0.1402	0.007741	440.00	0.0001277
0.9795	0.1390	0.007788	441.05	0.0001286
0.9790	0.1376	0.007835	442.90	0.0001293
0.9785	0.1364	0.007859	445.39	0.0001301
0.9780	0.1350	0.007906	447.38	0.0001308
0.9775	0.1338	0.007930	449.98	0.0001317
0.9770	0.1329	0.007977	450.50	0.0001323
0.9765	0.1317	0.008000	453.19	0.0001330
0.9760	0.1305	0.008024	455.94	0.0001339
0.9755	0.1296	0.008071	456.58	0.0001343
0.9750	0.1284	0.008095	459.43	0.0001352
0.9745	0.1272	0.008142	461.01	0.0001360
0.9740	0.1263	0.008166	463.11	0.0001367
0.9735	0.1256	0.008189	464.38	0.0001371
0.9730	0.1246	0.008236	465.22	0.0001378
0.9725	0.1237	0.00826	467.43	0.0001384
0.9720	0.1227	0.008284	469.68	0.0001391
0.9715	0.1218	0.008331	470.64	0.0001397
0.9710	0.1211	0.008354	472.06	0.0001404
0.9705	0.1201	0.008378	474.43	0.0001410
0.9700	0.1192	0.008425	475.51	0.0001415

(continued)

$F(\rho)$	C_a	C_b	C_σ	C_δ
0.969	0.1175	0.008472	479.50	0.0001428
0.968	0.1161	0.008520	482.66	0.0001441
0.967	0.1147	0.008567	485.93	0.0001452
0.966	0.1135	0.008614	488.29	0.0001463
0.965	0.1123	0.008661	490.73	0.0001474
0.964	0.1109	0.008708	494.30	0.0001482
0.963	0.1097	0.008756	496.92	0.0001491
0.962	0.1088	0.008826	497.21	0.0001502
0.961	0.1076	0.008874	499.99	0.0001511
0.960	0.1064	0.008921	502.86	0.0001519
0.959	0.1055	0.008968	504.69	0.0001530
0.958	0.1043	0.009015	507.73	0.0001539
0.957	0.1034	0.009062	509.70	0.0001546
0.956	0.1024	0.009110	511.73	0.0001554
0.955	0.1015	0.009157	513.83	0.0001563
0.954	0.1005	0.009204	515.99	0.0001572
0.953	0.09959	0.009228	519.55	0.0001581
0.952	0.09888	0.009275	520.61	0.0001589
0.951	0.09794	0.009298	524.29	0.0001596
0.950	0.09723	0.009346	525.44	0.0001604
0.948	0.09558	0.009416	530.50	0.0001622
0.946	0.09177	0.009511	533.14	0.0001635
0.944	0.09298	0.009582	535.91	0.0001650
0.942	0.09157	0.009652	540.21	0.0001663
0.940	0.09039	0.009723	543.28	0.0001676
0.938	0.08921	0.009794	546.48	0.0001692
0.936	0.08803	0.009865	549.83	0.0001705
0.934	0.08685	0.009912	554.65	0.0001720
0.932	0.08567	0.009983	558.30	0.0001731
0.930	0.08257	0.01005	560.55	0.0001742

(continued)

$F(\rho)$	C_a	C_b	C_σ	C_δ
0. 928	0. 08378	0. 01010	564. 21	0. 0001753
0. 926	0. 08284	0. 01017	566. 67	0. 0001764
0. 924	0. 08189	0. 01022	570. 56	0. 0001775
0. 922	0. 08095	0. 01029	573. 24	0. 0001785
0. 920	0. 08024	0. 01034	575. 66	0. 0001796
0. 918	0. 07930	0. 01041	578. 55	0. 0001807
0. 916	0. 07859	0. 01045	581. 12	0. 0001816
0. 914	0. 07788	0. 01050	583. 77	0. 0001827
0. 912	0. 07717	0. 01057	585. 18	0. 0001836
0. 910	0. 07623	0. 01062	589. 80	0. 0001846
0. 908	0. 07552	0. 01067	592. 69	0. 0001855
0. 906	0. 07481	0. 01071	595. 66	0. 0001866
0. 904	0. 07434	0. 01076	596. 82	0. 0001875
0. 902	0. 07363	0. 01083	598. 62	0. 0001884
0. 900	0. 07292	0. 01088	601. 81	0. 0001892
0. 895	0. 07151	0. 01100	607. 14	0. 0001914
0. 890	0. 07009	0. 01112	612. 83	0. 0001934
0. 885	0. 06891	0. 01123	616. 77	0. 0001953
0. 880	0. 06750	0. 01135	623. 17	0. 0001973
0. 875	0. 06655	0. 01145	626. 79	0. 0001988
0. 870	0. 06537	0. 01156	631. 60	0. 0002006
0. 865	0. 06419	0. 01166	638. 00	0. 0002023
0. 860	0. 06325	0. 01175	642. 32	0. 0002038
0. 855	0. 06230	0. 01185	646. 86	0. 0002056
0. 850	0. 06136	0. 01196	650. 33	0. 0002073
0. 840	0. 05971	0. 01215	657. 94	0. 0002099
0. 830	0. 05806	0. 01234	666. 31	0. 0002128
0. 820	0. 05664	0. 01251	673. 95	0. 0002154
0. 810	0. 05546	0. 01267	679. 32	0. 0002178
0. 800	0. 05428	0. 01284	685. 16	0. 0002204

(continued)

$F(\rho)$	C_a	C_b	C_σ	C_δ
0.75	0.04885	0.01362	717.75	0.0002306
0.70	0.04508	0.01433	739.43	0.0002389
0.65	0.04177	0.01503	760.33	0.0002459
0.60	0.03918	0.01567	777.75	0.0002516
0.55	0.03705	0.01628	791.35	0.0002564
0.50	0.03493	0.01694	806.73	0.0002609
0.45	0.03328	0.01758	816.10	0.0002644
0.40	0.03186	0.01820	823.62	0.0002677
0.35	0.03044	0.01879	834.86	0.0002701
0.30	0.02926	0.01945	839.01	0.0002723
0.25	0.02808	0.02006	847.52	0.0002742
0.20	0.02714	0.02074	848.07	0.0002758
0.15	0.02620	0.02143	850.57	0.0002764
0.10	0.02525	0.02214	854.14	0.0002773
0.05	0.02431	0.02287	857.27	0.0002780
0.00	0.0236	0.0236	857.27	0.0002782

From the aforementioned, we could summarize the calculaton procedure for the point contact elastic deformation and contacting stress as follows:

- 1) Determine the principle curvatures $\rho_{I1}, \rho_{I2}, \rho_{II1}, \rho_{II2}$.
- 2) Calculate $\sum \rho$.
- 3) Calculate $F(\rho)$.
- 4) From Table 2-4, get the factors $C_a, C_b, C_\sigma, C_\delta$ values.
- 5) From the equations (2-30) to (2-33), calculate $a, b, \sigma_{\max}, \delta$ values.

[**Example 2-5**] To calculate the maximum contact stress $\sigma_{\max}$ and a, b, δ in the bearing 7208 C. The design factors are the same as in Example 2-4.

[**Resolve**] $\sum \rho$ and $F(\rho)$ of the bearing 7208 C can be got from Example 2-4. According to the equations (2-30) to (2-33) and checking out the related factors from Table 2-4, the calculation results are as follows:

(1) On the contact point of ball with inner ring raceway

$$\Sigma \rho_i = 0.203 \quad F(\rho_i) = 0.955$$

$$\mu = 4.30 \quad \nu = 0.388$$

$$\sqrt[3]{\frac{1}{\Sigma \rho_i}} = \sqrt[3]{\frac{1}{0.203}} = 1.7015$$

$$\sqrt[3]{(\Sigma \rho_i)^2} = \sqrt[3]{0.203^2} = 0.3454$$

$$\sqrt[3]{\Sigma \rho_i} = \sqrt[3]{0.203} = 0.5877$$

$$C_a = 0.1015$$

$$C_b = 0.00916$$

$$C_\sigma = 513.83$$

$$C_\delta = 0.000156$$

$$a = C_a \cdot \sqrt[3]{\frac{Q}{\Sigma \rho_i}} = 0.173 Q^{1/3} \text{ mm}$$

$$b = C_b \cdot \sqrt[3]{\frac{Q}{\Sigma \rho_i}} = 0.0156 Q^{1/3} \text{ mm}$$

$$\delta = C_\delta \sqrt[3]{Q^2 \Sigma \rho_i} = 9.2 * 10^{-5} Q^{2/3} \text{ mm}$$

$$\sigma_{\max} = C_\delta \sqrt[3]{Q (\Sigma \rho_i)^2} = 177.48 Q^{1/3} \text{ N/mm}^2$$

(2) On the contact point of ball with outer ring raceway

$$\Sigma \rho_e = 0.135 \quad F(\rho_e) = 0.932$$

$$\mu = 3.63 \quad \nu = 0.423$$

$$\sqrt[3]{\frac{1}{\Sigma \rho_e}} = \sqrt[3]{\frac{1}{0.135}} = 1.949$$

$$\sqrt[3]{(\Sigma \rho_e)^2} = \sqrt[3]{0.135^2} = 0.263$$

$$\sqrt[3]{\Sigma \rho_e} = \sqrt[3]{0.135} = 0.513$$

$$C_a = 0.0857$$

$$C_b = 0.00998$$

$$C_\sigma = 558.30$$

$$C_\delta = 0.000173$$

$$a = C_a \sqrt[3]{\frac{Q}{\Sigma \rho_e}} = 0.167 Q^{1/3} \text{ mm}$$

$$b = C_b \sqrt[3]{\frac{Q}{\Sigma \rho_e}} = 0.0195 Q^{1/3} \text{ mm}$$

$$\delta = C_{\delta} \sqrt[3]{Q^2 \sum \rho_e} = 8.87 * 10^{-5} Q^{2/3} \text{ mm}$$

$$\sigma_{\max} = C_{\sigma} \sqrt[3]{Q (\sum \rho_e)^2} = 146.83 Q^{1/3} \text{ N/mm}^2$$

2.2.3 Line Contact

In line contact condition, for steel making bearing, the contact area width $2b$, the maximum contact stress $\sigma_{\max}$ and the elastic approach quantity δ can be approximately calculated as follows:

$$(1) \quad b = \sqrt{\frac{8 \left(1 - \frac{1}{m^2}\right)}{\pi E} \frac{Q}{L_{\text{we}} \sum \rho}} = \sqrt{\frac{8}{\pi E}} \sqrt{\frac{Q}{L_{\text{we}} \sum \rho}}$$

Putting the steel parameters $E = 2.079 * 10^5 \text{ N/mm}^2$, $m = \frac{10}{3}$, into the above equation, we can

$$\text{get:} \quad b = 0.00334 \sqrt{\frac{Q}{L_{\text{we}} \sum \rho}} \text{ mm} \quad (2-34)$$

$$(2) \quad \sigma_{\max} = \frac{4Q}{\pi 2b L_{\text{we}}} = \frac{4Q}{2\pi L_{\text{we}}} \frac{\sqrt{\pi E L_{\text{we}} \sum \rho}}{\sqrt{8Q}} = \sqrt{\frac{E}{2\pi}} \sqrt{\frac{Q \sum \rho}{L_{\text{we}}}}$$

Putting the steel parameters $E = 2.079 * 10^5 \text{ N/mm}^2$, $m = \frac{10}{3}$, into the above equation, we can get:

$$\sigma_{\max} = 190.6 \sqrt{\frac{Q \sum \rho}{L_{\text{we}}}} \text{ N/mm}^2 \quad (2-35)$$

$$(3) \quad \delta = 3.84 * 10^{-5} \frac{Q^{0.9}}{L_{\text{we}}^{0.8}} \text{ mm} \quad (2-36)$$

In the condition of two contacting tapered parts having the same tapered top, the contact area is a ladder-shaped area. In this situation, the semi-width b is designated by the half of the average width of the ladder-shaped area.

The elastic approach quantity δ is designated as the distance that the cylinder axis aparts from one point of a supporting plane when a cylinder with limited length is pressed by two infinite large plane surfaces, while the three contacting bodies are made by the same material. These conditions are nearly similar to the cylindrical roller bearing contacting condition, and the

equation (2 - 36) could be approximately used for the cylindrical roller bearing calculation.

The calculation equations of a , b , $\sigma_{\max}$, δ values for every steel making rolling bearing are listed in Table 2 - 5.

2.2.4 Maximum Shear Stress $\tau_{\max}$

In rolling bearing, the subsurface shear stress is the decisive factor for the contact fatigue failure. This subsurface shear stress is in the plane parallel to the contact surface and below the contact surface. During the passage of a loaded rolling element rolling over a point on the raceway surface, the maximum shear stress varies between $+\tau_{\max}$ to $-\tau_{\max}$, as shown in Figure 2 - 14 and Figure 2 - 15. If using T expresses the relative total shear stress, we can get:

$$T = \frac{2\tau_{\max}}{\sigma_{\max}} = \frac{\sqrt{2t-1}}{t(t+1)} \tag{2-37}$$

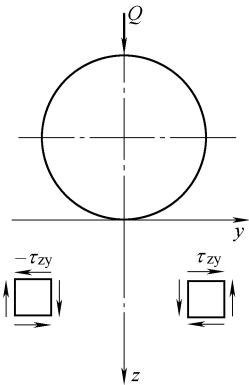


Figure 2 - 14 shear stress τ_{xy} direction

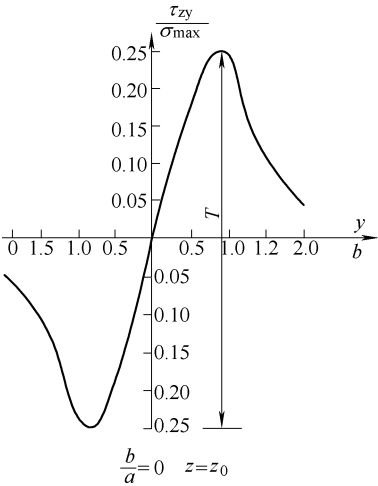
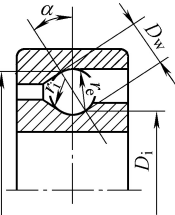
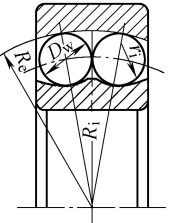
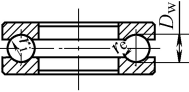


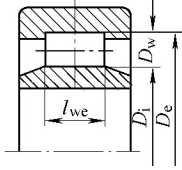
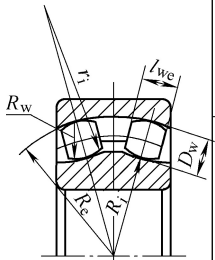
Figure 2 - 15 Shear stress extent

In the equation, t is an auxiliary factor, which can be decided by the following equation:

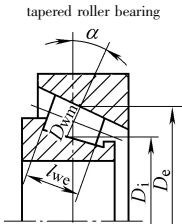
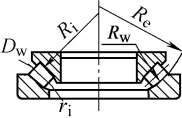
Table 2-5 Steel making rolling bearing a , b , $\sigma_{\max}$, δ equations

Bearing type		$\Sigma\rho$	$F(\rho)$	a	b	$\sigma_{\max}$	δ	k	note
deep groove and angular contact ball bearing 	rolling element contact point with inner ring race- way	$\frac{1}{D_w}\left(4 + \frac{2\gamma}{1-\gamma} - \frac{1}{f_i}\right)$	$\frac{\frac{1}{f_i} + \frac{2\gamma}{1-\gamma}}{4 + \frac{2\gamma}{1-\gamma} - \frac{1}{f_i}}$	$C_a\sqrt[3]{\frac{Q}{\Sigma\rho_i}}$	$C_b\sqrt[3]{\frac{Q}{\Sigma\rho_i}}$	$C_\sigma\sqrt[3]{Q(\Sigma\rho_i)^2}$	$C_\delta\sqrt[3]{Q^2\Sigma\rho_i}$	$C_{\delta i}\sqrt[3]{\Sigma\rho_i}$ + $C_{\delta e}\sqrt[3]{\Sigma\rho_e}$	$f_i = \frac{\gamma_i}{D_w}$ $f_e = \frac{\gamma_e}{D_w}$ $\gamma = \frac{D_w \cos\alpha}{D_{pw}}$ deep groove ball bearing $\alpha = 0$
	rolling element contact point with outer ring race- way	$\frac{1}{D_w}\left(4 - \frac{2\gamma}{1+\gamma} - \frac{1}{f_e}\right)$	$\frac{\frac{1}{f_e} - \frac{2\gamma}{1+\gamma}}{4 - \frac{2\gamma}{1+\gamma} - \frac{1}{f_e}}$	$C_a\sqrt[3]{\frac{Q}{\Sigma\rho_e}}$	$C_b\sqrt[3]{\frac{Q}{\Sigma\rho_e}}$	$C_\sigma\sqrt[3]{Q(\Sigma\rho_e)^2}$	$C_\delta\sqrt[3]{Q^2\Sigma\rho_e}$		
aligning ball bearing 	rolling element contact point with inner ring race- way	$\frac{4}{D_w} + \frac{1}{R_i} - \frac{1}{\gamma_i}$	$\frac{\frac{1}{R_i} + \frac{1}{\gamma_i}}{\frac{4}{D_w} + \frac{1}{R_i} - \frac{1}{\gamma_i}}$	$C_a\sqrt[3]{\frac{Q}{\Sigma\rho_i}}$	$C_b\sqrt[3]{\frac{Q}{\Sigma\rho_i}}$	$C_\sigma\sqrt[3]{Q(\Sigma\rho_i)^2}$	$C_\delta\sqrt[3]{Q^2\Sigma\rho_i}$		
	rolling element contact point with outer ring race- way	$\frac{4}{D_w} - \frac{2}{R_e}$	0	$0.0236\sqrt[3]{\frac{Q}{\Sigma\rho_e}}$	$0.0236\sqrt[3]{\frac{Q}{\Sigma\rho_e}}$				
thrust ball bearing 	rolling element contact point with shaft washer race- way	$\frac{1}{D_w}\left(4 - \frac{1}{f_i}\right)$	$\frac{1}{4f_i - 1}$	$C_a\sqrt[3]{\frac{Q}{\Sigma\rho_i}}$	$C_b\sqrt[3]{\frac{Q}{\Sigma\rho_i}}$	$C_\sigma\sqrt[3]{Q(\Sigma\rho_i)^2}$	$C_\delta\sqrt[3]{Q^2\Sigma\rho_i}$	$C_{\delta i}\sqrt[3]{\Sigma\rho_i}$ + $C_{\delta e}\sqrt[3]{\Sigma\rho_e}$	$f_i = \frac{\gamma_i}{D_w}$ $f_e = \frac{\gamma_e}{D_w}$

(continued)

Bearing type		$\Sigma\rho$	$F(\rho)$	a	b	$\sigma_{\max}$	δ	k	note
	rolling element contact point with housing washer raceway	$\frac{1}{D_w}\left(4 - \frac{1}{f_e}\right)$	$\frac{1}{4f_e - 1}$	$C_\alpha\sqrt[3]{\frac{Q}{\Sigma\rho_e}}$	$C_\psi\sqrt[3]{\frac{Q}{\Sigma\rho_e}}$	$C_\alpha\sqrt[3]{Q(\Sigma\rho_e)^2}$	$C_\delta\sqrt[3]{Q^2\Sigma\rho_e}$	$C_{\delta i}\sqrt[3]{\Sigma\rho_i}$ + $C_{\delta e}\sqrt[3]{\Sigma\rho_e}$	$f_i = \frac{\gamma_i}{D_w}$ $f_e = \frac{\gamma_e}{D_w}$
cylindrical roller bearing 	rolling element contact point with inner ring raceway	$\frac{2}{D_w} + \frac{2}{D_i}$			$0.00334\sqrt[3]{\frac{Q}{L_{wei}\Sigma\rho_i}}$	$19\sqrt[3]{\frac{Q}{L_{wei}\Sigma\rho_i}}$	$3.84 \times 10^{-5}\frac{Q^{0.9}}{L_{wi}^{0.8}}$	$\frac{3.84 \times 10^{-5}}{L_{wei}^{0.8}}$ + $\frac{3.84 \times 10^{-5}}{L_{wee}^{0.8}}$	If roller or raceway are crown shape, we can calculate according to point contact equations.
	rolling element contact point with outer ring raceway	$\frac{2}{D_w} - \frac{2}{D_e}$			$0.00334\sqrt[3]{\frac{Q}{L_{wee}\Sigma\rho_e}}$	$191\sqrt[3]{\frac{Q}{L_{wee}\Sigma\rho_e}}$	$3.84 \times 10^{-5}\frac{Q^{0.9}}{L_{wee}^{0.8}}$		
aligning roller bearing 	roller element contact point with inner ring raceway	$\frac{2}{D_w} + \frac{1}{R_w}$ + $\frac{1}{R_i} - \frac{1}{r_i}$	$\frac{2}{D_w} - \frac{1}{R_w} + \frac{1}{R_i} + \frac{1}{r_i}$ $\frac{2}{D_w} + \frac{1}{R_w} + \frac{1}{R_i} - \frac{1}{r_i}$	$C_\alpha\sqrt[3]{\frac{Q}{\Sigma\rho_i}}$	$C_\psi\sqrt[3]{\frac{Q}{\Sigma\rho_i}}$	$C_\alpha\sqrt[3]{Q(\Sigma\rho_i)^2}$	$C_\delta\sqrt[3]{Q^2\Sigma\rho_i}$	$C_{\delta i}\sqrt[3]{\Sigma\rho_i}$ + $C_{\delta e}\sqrt[3]{\Sigma\rho_e}$	
	rolling element contact point with outer ring raceway	$\frac{2}{D_w} + \frac{1}{R_w}$ - $\frac{2}{R_e}$	$\frac{2}{D_w} - \frac{1}{R_w}$ $\frac{2}{D_w} + \frac{1}{R_w} - \frac{2}{R_e}$	$C_\alpha\sqrt[3]{\frac{Q}{\Sigma\rho_e}}$	$C_\psi\sqrt[3]{\frac{Q}{\Sigma\rho_e}}$	$C_\alpha\sqrt[3]{Q(\Sigma\rho_e)^2}$	$C_\delta\sqrt[3]{Q^2\Sigma\rho_e}$		

(continued)

Bearing type		$\Sigma \rho$	$F(\rho)$	a	b	$\delta_{\max}$	δ	k	note
	roller element contact point with inner ring raceway	$\frac{2}{D_{wm}} + \frac{2 \cos \alpha_i}{D_i}$			$0.00334 \left(\sqrt{\frac{Q}{L_{wei} \Sigma \rho_i}} \right) 191 \left(\sqrt{\frac{Q}{L_{wei} \Sigma \rho_i}} \right)$	$3.84 \times 10^{-5} \frac{Q^{0.9}}{L_{wei}^{0.8}}$		$\frac{3.84 \times 10^{-5}}{L_{wei}^{0.8}} + \frac{3.84 \times 10^{-5}}{L_{wee}^{0.8}}$	If the cone angle is $2^\circ \sim 4^\circ$ in tapered bearing calculating general should use average diameter. If roller or raceway is crown shape, calculation should use point contact equations.
	rolling element contact point with outer ring raceway	$\frac{2}{D_{wm}} - \frac{2 \cos \alpha}{D_e}$			$0.00334 \left(\sqrt{\frac{Q}{L_{wee} \Sigma \rho_e}} \right) 191 \left(\sqrt{\frac{Q}{L_{wee} \Sigma \rho_e}} \right)$	$3.84 \times 10^{-5} \frac{Q^{0.9}}{L_{wee}^{0.8}}$			
	roller element contact point with inner ring raceway	$\frac{2}{D_w} + \frac{1}{R_w} + \frac{1}{R_i} - \frac{1}{r_i}$	$\frac{2}{D_w} - \frac{1}{R_w} + \frac{1}{R_i} + \frac{1}{r_i}$	$C_{\alpha} \left(\sqrt[3]{\frac{Q}{\Sigma \rho_i}} \right)$	$C_{\beta} \left(\sqrt[3]{\frac{Q}{\Sigma \rho_i}} \right)$	$C_{\alpha} \left(\sqrt[3]{Q (\Sigma \rho_i)^2} \right)$	$C_{\beta} \left(\sqrt[3]{Q^2 \Sigma \rho_i} \right)$	$C_{\beta} \left(\sqrt[3]{\Sigma \rho_i} \right) + C_{\beta e} \left(\sqrt[3]{\Sigma \rho_e} \right)$	
	rolling element contact point with outer ring raceway	$\frac{2}{D_w} + \frac{1}{R_w} - \frac{2}{R_e}$	$\frac{2}{D_w} - \frac{1}{R_w} + \frac{1}{R_e}$	$C_{\alpha} \left(\sqrt[3]{\frac{Q}{\Sigma \rho_e}} \right)$	$C_{\beta} \left(\sqrt[3]{\frac{Q}{\Sigma \rho_e}} \right)$	$C_{\alpha} \left(\sqrt[3]{Q (\Sigma \rho_e)^2} \right)$	$C_{\beta} \left(\sqrt[3]{Q^2 \Sigma \rho_e} \right)$		

$$\frac{b}{a} = \sqrt{(t^2 - 1)(2t - 1)} \quad (2-38)$$

The maximum shear stress $\tau_{\max}$ is at the z_0 depth, below the contact surface.

$$z_0 = \zeta b \quad (2-39)$$

$$\text{Where} \quad \zeta = \frac{1}{(t+1)\sqrt{2t-1}} \quad (2-40)$$

Obviously T and ζ are the function of a and b . Their relations are listed in Table 2-6.

Table 2-6 Relative shear stress T

t	1	1.02	1.05	1.1	1.2	1.2808
b/a	0	0.2050	0.3358	0.5020	0.7849	1
T	0.5000	0.4950	0.4872	0.4742	0.4482	0.4278
z_0/b	0.5000	0.4854	0.4651	0.4347	0.3842	0.3509

Some important rolling bearings have to be made by special material and through the carburization treatment. Generally, the carburization depth should be greater than the z_0 , usually taking a safety factor.

When calculating the maximum shear stress $\tau_{\max}$ and z_0 , first we should calculate a and b , then checking out the relative T and z_0/b from Table 2-6.

2.2.5 Elastic Deformation Constant k

The elastic approach quantity δ calculated from the equations (2-29) and (2-36) is the elastic deformation value of one contact point of rolling element with one ring raceway. The total elastic deformation value of one rolling element contacting with inner ring and outer ring should be:

$$\delta_n = \delta_i + \delta_e \quad (2-41)$$

Where

δ_i — elastic approach quantity of one contact point of rolling element with inner ring raceway;

δ_e — elastic approach quantity of one contact point of rolling element

with outer ring raceway;

δ_n — total elastic deformation value of one rolling element contacting with inner ring and outer ring.

For point contact, from the equation (2 - 29) we can get:

$$\delta_n = C_{\delta i} C_E^{-1} \sqrt[3]{Q^2 \Sigma \rho_i} + C_{\delta e} C_E^{-1} \sqrt[3]{Q^2 \Sigma \rho_e} = K Q^{2/3} \quad (2 - 42)$$

$$K = C_{\delta i} C_E^{-1} \sqrt[3]{\Sigma \rho_i} + C_{\delta e} C_E^{-1} \sqrt[3]{\Sigma \rho_e} \quad (2 - 43)$$

The calculation equations of elastic deformation constant K for each type of rolling bearings are listed in Table 2 - 5. For a given rolling bearing, because the principle curvature is a known value, K is a constant.

In ball bearings, because the curvature in the principle plane perpendicular to the axial plane is very small, if omitting the curvature in this principle plane, the equation can be simplified.

Equation (2 - 18) is simplified as

$$\Sigma \rho \approx \frac{1}{D_w} \left(4 - \frac{1}{f} \right) \quad (2 - 44)$$

Equation (2 - 20) is simplified as

$$F(\rho) \approx \frac{\frac{1}{f}}{4 - \frac{1}{f}} \quad (2 - 45)$$

For steel making rolling bearings, if the raceway curvature factors of inner ring and outer ring are the same, the equation (2 - 43) can be simplified as:

$$K = 2C_{\delta} \sqrt[3]{4 - \frac{1}{f}} \frac{1}{D_w^{1/3}} = k \frac{1}{D_w^{1/3}} \quad (2 - 46)$$

Where

$$k = 2C_{\delta} \sqrt[3]{4 - \frac{1}{f}} \quad (2 - 47)$$

If the f value is known, then from Equation (2 - 45) we can get $F(\rho)$, and can check out the relative C_{δ} for $F(\rho)$ from Table 2 - 2, From Equation (2 - 47) we can calculate the k value.

From different f values we can calculate the k values listed in Table 2 - 7 and Figure 2 - 16.

Table 2-7 *k* values

<i>f</i>	0.515	0.5175	0.52	0.525	0.53	0.54
<i>k</i>	0.000422	0.000437	0.000450	0.000473	0.000493	0.000525
<i>f</i>	0.55	0.56	0.57	0.58	0.59	0.60
<i>k</i>	0.000549	0.000571	0.000588	0.000603	0.000616	0.000628

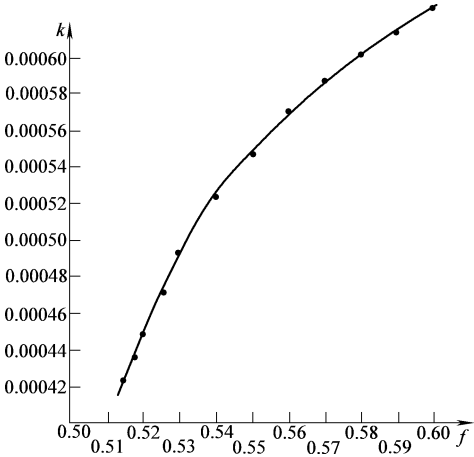


Figure 2-16 Relating Figure for *k* and *f*

For line contact, for steel making bearing, from Equation (2-36) we can get:

$$\delta_n = 3.84 * 10^{-5} \left(\frac{1}{L_{wei}^{0.8}} + \frac{1}{L_{wee}^{0.8}} \right) Q^{0.9} = KQ^{0.9} \tag{2-48}$$

Where
$$K = 3.84 * 10^{-5} \left(\frac{1}{L_{wei}^{0.8}} + \frac{1}{L_{wee}^{0.8}} \right) \tag{2-49}$$

In Equations (2-48) and (2-49), the unit of *Q* is N.

2.2.6 Radial Deflection δ_r and Axial Deflection δ_a in Rolling Bearings

(1) Load angle

The load applied to a bearing could be divided as two components. Those are the axial load F_a and radial load F_r .

The axial load F_a designates the load component that passes through the

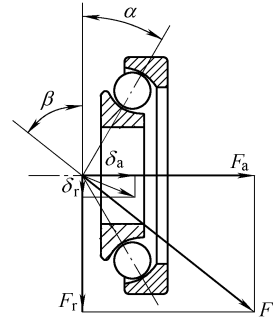
load acting center and along with the bearing center axial line.

The radial load F_r designates the load component that passes through the load acting center and in the plane parallel with bearing radial plane.

The angle β between the resulting force vector F (of the axial load F_a and the radial load F_r) and the radial plane is called the load angle, as shown in Figure 2-17.

From Figure 2-17 we can get:

$$\tan\beta = \frac{F_a}{F_r} \quad (2-50) \quad \text{Figure 2-17 Load angle } \beta$$



(2) Relationship between load angle and bearing deflection

Because of the deformation between rolling element and ring raceways, while subjecting to load, the bearing inner ring center will move some distance relative to the outer ring center. As shown in Figure 2-17, under the axial load F_a and radial load F_r , the bearing inner ring and outer ring will move δ_a and δ_r relative to each other. Generally, the relative moving direction (that is the resulting vector of δ_a and δ_r) of the bearing rings is not overlapped with the resulting direction of F_a and F_r .

If the contact angle keeps not changing, according to the theory of load distribution in rolling bearings (see Chapter 2.3), we can calculate the ratio of δ_a/δ_r .

Figure 2-18 is the relationship of $\delta_a \tan\alpha/\delta_r$ and $\tan\alpha/\tan\beta$. In this figure the solid line designates the point contact, and the dotted line designates the line contact.

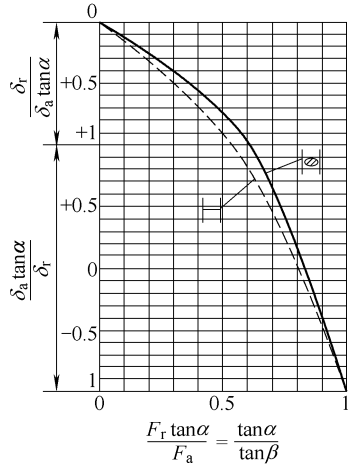


Figure 2-18 Relationship of $\delta_a \tan\alpha/\delta_r$ and $\tan\alpha/\tan\beta$

In angular contact ball bearings, the inner ring and outer ring are separable.

For the different relative position of inner ring with outer ring, the load distribution is different. So there is a hypothesis that if the bearing is subjected to load just by half of the ring raceways, the $\delta_a = 0$. If the bearing is subjected to load by the ring raceways larger than half of the raceways, then $\delta_a > 0$. If the bearing is subjected to load by the ring raceways smaller than half of the raceways, then $\delta_a < 0$, which means the inner ring and outer ring intend to separate.

In Figure 2-18, the plus sign (+) expresses that the δ_a direction makes the inner ring and outer ring pressed tightly each other, while the negative sign (—) expresses that the δ_a direction makes the inner ring and outer ring separated each other.

(3) Pure radial displacement and pure axial displacement

If the inner ring and the outer ring have translationally moved a relative δ_a and δ_r , according to the theory of load distribution in rolling bearings (see Chapter 2.3), then the total elastic contact deformation along the normal line on the maximum subjected load contacting position is:

$$\delta_{\max} = \delta_a \sin\alpha + \delta_r \cos\alpha \quad (2-51)$$

If there is only a pure radial displacement, then the $\delta_a = 0$. From the equation (2-51) we can get:

$$\delta_r = \frac{\delta_{\max}}{\cos\alpha} \quad (2-52)$$

If there is only a pure axial displacement, then the $\delta_r = 0$. From Equation (2-51) we can get:

$$\delta_a = \frac{\delta_{\max}}{\sin\alpha} \quad (2-53)$$

Putting the equations (2-42) and (2-48) into the equations (2-52) and (2-53), we can get:

For point contact:

$$\delta_r = \frac{k}{\cos\alpha} Q_{\max}^{2/3} \quad (2-54)$$

$$\delta_a = \frac{k}{\sin \alpha} Q_{\max}^{2/3} \quad (2-55)$$

$$\text{For line contact} \quad \delta_r = \frac{k}{\cos \alpha} Q_{\max}^{0.9} \quad (2-56)$$

$$\delta_a = \frac{k}{\sin \alpha} Q_{\max}^{0.9} \quad (2-57)$$

Here in the equations, k is the elastic deformation constant, which can be obtained from the equations (2-43) and (2-49).

The k values of the standard design of every steel making rolling bearings can be checked out from Table 2-8.

Table 2-8 k values of the standard design of every steel making rolling bearings

Bearing type	k
Deep groove and angular contact ball bearings	$0.00044/D_w^{1/3}$
Aligning ball bearings	$0.00069/D_w^{1/3}$
Thrust ball bearings	$0.00052/D_w^{1/3}$
Tapered roller bearings	$0.000077/l_{we}^{0.8}$
Cylindrical roller bearings	
Thrust roller bearings	

In Equations (2-54) to (2-57), the unit of $Q_{\max}$ is N.

2.3 Load Distribution in Rolling Bearings

The subjecting load on rolling bearing is transferred from one ring to another ring through rolling elements. So the determining factor for rolling bearing load capacity is the rolling element load value.

When there are more than two rolling elements subjecting to load (usually the rolling bearing is this condition), this system is a static-indeterminate problem. In order to determine the load on every rolling element, it is bound to consider the deformations between rolling element and ring raceways.

2.3.1 Load Distribution in Radial Contact Rolling Bearings

In order to analyze this problem simply, first of all, we make some hypotheses:

- 1) Rolling bearing is subjected only to radial load F_r .
- 2) The geometry shapes of rolling bearing parts are ideal correct shapes.
- 3) The deformations of rolling element and raceways are in the elastic range.
- 4) The bearing radial clearance is zero, i. e. $G_r = 0$.

After the radial bearing subjected to the radial load F_r , the upper half circle of the rolling elements is not subjected to any load, and only the lower half circle of the parts is subjected to load. Because of the elastic deformation between rolling elements and ring raceways, the inner ring center moves downward δ_r relative to the outer ring center. In this condition, each rolling element contact point elastic deformation is different, as shown in Figure 2-19.

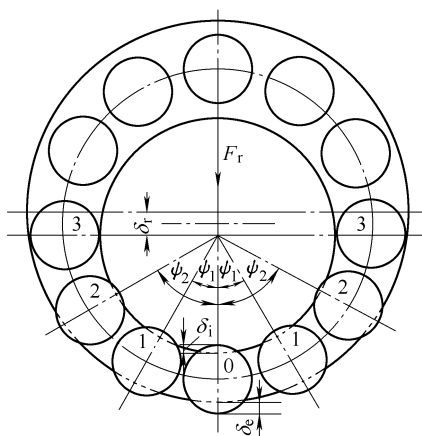


Figure 2-19 Elastic deformation in radial bearing

In Figure 2-19:

ψ — The angle between each rolling element center and the maximum

load element, $\psi_0 = 0^\circ$, $\psi_1 = \frac{360^\circ}{z}$, $\psi_2 = 2 * \frac{360^\circ}{z}$,

$\psi_3 = 3 * \frac{360^\circ}{z} \dots$;

δ_i — The elastic deformation between the inner ring raceway and rolling element;

δ_e — The elastic deformation between the outer ring raceway and rolling

element.

Knowing from Section 2.2, the total elastic deformation^① between the inner ring raceway and outer ring raceway is as:

$$\delta = KQ^n \quad (2-58)$$

For ball bearing $n = 2/3$

For roller bearing $n = 0.9$

As shown in Figure 2-20, under the radial load F_r , each point on the inner ring raceway moves down δ_r distance. The contact deformation value designates the deformation value along the vertical line direction, and the elastic deformation between the inner ring raceway and outer ring raceway with rolling elements should be:

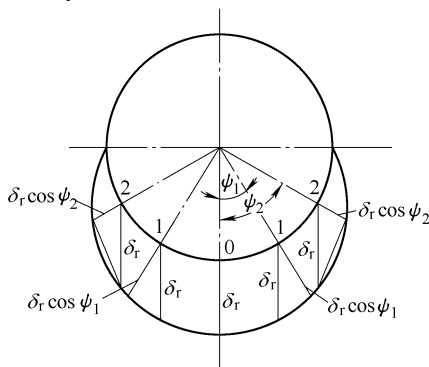


Figure 2-20 Radial deflection
in radial bearing

$$\delta_\psi = \delta_r \cos \psi = \delta_{\max} \cos \psi \quad (2-59)$$

Here δ_ψ — The elastic deformation value at the ψ angle position aparting from the maximum load rolling element;

$\delta_{\max}$ — The elastic deformation value at the position of the maximum load rolling element.

From the equation (2-59), we can get:

$$\frac{\delta_\psi}{\delta_{\max}} = \cos \psi \quad (2-60)$$

From the equations (2-58) and (2-60) we can get:

$$\frac{Q_\psi}{Q_{\max}} = \left(\frac{\delta_\psi}{\delta_{\max}} \right)^{1/n} = \cos^{1/n} \psi \quad (2-61)$$

In these equations:

① For simplifying, δ expresses the total deformation Value between the rolling element and the inner ring and outer ring, omitting the corner note n in the equations (2-42) and (2-48).

Q_ψ — The rolling element load value at the ψ angle position aparting from the maximum load rolling element;

$Q_{\max}$ — The maximum rolling element load.

For ball bearing $t = 3/2$;

For roller bearing $t = 1$.

Figure 2-21 shows the bearing load distribution. Each rolling element load can be resolved into two components. From the force balance, we can get:

$$F_r = Q_{\max} + 2Q_1 \cos \psi_1 + 2Q_2 \cos \psi_2 + \dots \quad (2-62)$$

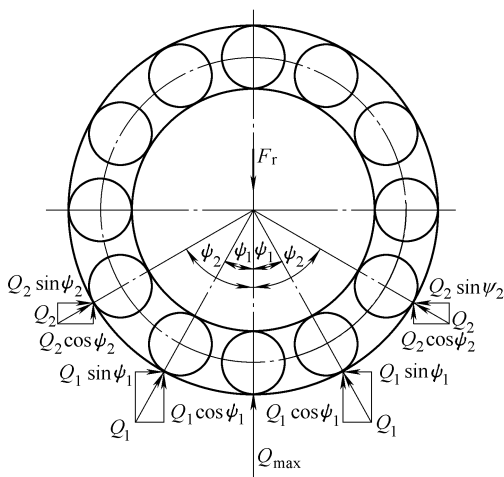


Figure 2-21 Radial bearing load distribution

If putting the equation (2-61) into the equation (2-62), we can get:

$$Q_{\max} = \frac{F_r}{1 + 2 \cos^{t+1} \psi_1 + 2 \cos^{t+1} \psi_2 + \dots} \quad (2-63)$$

From the above equation, $Q_{\max}$ is related to the the subjecting load F_r and the rolling element numbers z .

If introducing:

$$J_r = \frac{1 + 2 (\cos^{t+1} \psi_1 + \cos^{t+1} \psi_2 + \dots)}{z} \quad (2-64)$$

Then the equation (2-63) can be rewritten as:

$$Q_{\max} = \frac{F_r}{zJ_r} \quad (2-65)$$

For different z value, the calculated J_r value are listed in Table 2-9.

Table 2-9 $1/J_r$ value

rolling element number z		7	8	10	12	15	20
$\frac{1}{J_r}$	ball bearing	4.34	4.35	4.34	4.36	4.37	4.36
	roller bearing	4.01	4.07	4.10	4.08	4.07	4.08

From Table 2-9, along with the increasing of z , the $1/J_r$ value intends nearly to be a constant. When a radial bearing is only subjected to radial load, the maximum rolling element load can be calculated from the following equation:

For ball bearing:

$$Q_{\max} = \frac{4.37F_r}{z} \quad (2-66)$$

For roller bearing:

$$Q_{\max} = \frac{4.08F_r}{z} \quad (2-67)$$

[**Example 2-6**] The deep groove ball bearing 6208 is subjected to radial load $F_r = 2.94\text{kN}$, steel ball number is $z = 9$, radial clearance is $G_r = 0$, Calculate the subjecting load on each ball.

[**Resolve**] From the equation (2-66) we can get:

$$Q_{\max} = \frac{4.37 * 2.94}{9} = 1.43\text{kN}$$

$$\psi_1 = \frac{360^\circ}{9} = 40^\circ, \quad \cos 40^\circ = 0.766$$

From the equation (2-61) we can get:

$$Q_1 = 1.43 * (\cos 40^\circ)^{1.5} \text{kN} = 0.96\text{kN}$$

$$Q_2 = 1.43 * (\cos 80^\circ)^{1.5} \text{kN} = 0.103\text{kN}$$

The subjecting load on different position of the ball is listed in Table (2-11).

2.3.2 Effect of Radial Clearance on Load Distribution in Radial Contact Rolling Bearings

Obviously, if there is a radial clearance G_r in a rolling bearing, the subjecting load region in rolling bearing would be reduced, and the maximum rolling element load would be increased. If under the radial load F_r , the bearing inner ring would even move a distance δ_r . If using G_r expresses the bearing radial clearance, we can get:

$$\delta_r = \delta_{\max} + \frac{G_r}{2} \quad (2-68)$$

Then:

$$\delta_\psi = \delta_r \cos\psi - \frac{G_r}{2} = \left(\delta_{\max} + \frac{G_r}{2} \right) \cos\psi - \frac{G_r}{2} \quad (2-69)$$

If introducing:

$$T = \frac{1}{2} \left(1 - \frac{G_r}{2\delta_{\max} + G_r} \right) \quad (2-70)$$

In this equation, T is the radial bearing load distribution factor. If $T = 0.5$, then $G_r = 0$; if $T < 0.5$, then $G_r > 0$; if $T > 0.5$, there is a negative radial clearance, that is $G_r < 0$, and the bearing is on preloading condition.

If putting the equation (2-70) into the equation (2-69), we can get:

$$\frac{\delta_\psi}{\delta_{\max}} = \left[1 - \frac{1}{2T} (1 - \cos\psi) \right]$$

From the equation (2-58) we can deduce that

$$Q_\psi = Q_{\max} \left[1 - \frac{1}{2T} (1 - \cos\psi) \right]^t \quad (2-71)$$

For ball bearing: $t = 3/2$;

For roller bearing: $t = 1$.

From the above equation, the subjecting load region is:

$$\cos\psi_0 = 1 - 2T \quad (2-72)$$

If the calculated result is $\psi_0 < 360^\circ/z$, this condition means that there is only one rolling element subjecting to load.

From the balance of the bearing subjecting forces, we can get:

$$F_r = \sum Q_\psi \cos\psi = zQ_{\max} \frac{\sum \left[1 - \frac{1}{2T} (1 - \cos\psi) \right]^t \cos\psi}{z}$$

Introducing the load distribution integral J_r

$$\begin{aligned} J_r &= \frac{1}{2\pi} \int_{-\psi_0}^{+\psi_0} \left[1 - \frac{1}{2T} (1 - \cos\psi) \right]^t \cos\psi d\psi \\ &\approx \frac{\sum \left[1 - \frac{1}{2T} (1 - \cos\psi) \right]^t \cos\psi}{z} \end{aligned}$$

We can get the same equation as the equation (2-65):

$$Q_{\max} = \frac{F_r}{zJ_r}$$

The J_r values are listed in Table 2-10.

Table 2-10 Radial bearing J_r values

T	J_r	
	ball bearing	roller bearing
0	$\frac{1}{z}$	$\frac{1}{z}$
0.05	0.0828	0.0910
0.10	0.1156	0.1268
0.15	0.1397	0.1528
0.20	0.1590	0.1736
0.25	0.1753	0.1909
0.30	0.1892	0.2055
0.35	0.2012	0.2179
0.40	0.2117	0.2286
0.45	0.2209	0.2376
0.50	0.2288	0.2453

The relationship of J_r with the T (the radial bearing load distribution factor) also can be checked out from Figure 2-22.

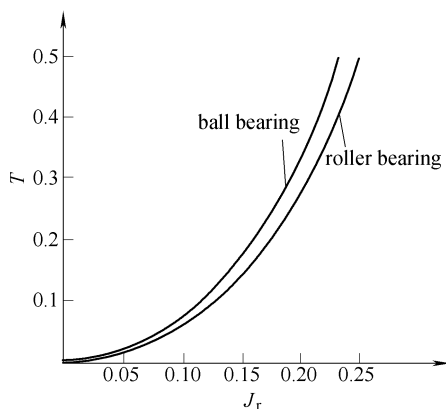


Figure 2 – 22 Radial bearing J_r values

From the equation (2 – 70), the load distribution factor T values are related to the radial clearance and the maximum elastic deformation $\delta_{\max}$ between rolling element and ring raceway. If knowing the radial load F_r and the radial clearance G_r , then we can calculate the subjecting load on each rolling element. The procedures are as follows:

First of all, we decide the bearing elastic deformation constant K . Secondly, we calculate the maximum elastic deformation $\delta_{\max}$ of the rolling element in the bearing. Then from the radial clearance G_r value we calculate the the subjecting load region distribution angle. At last, we calculate the subjecting load on every rolling element. Next using Example 2 – 7 describes the determining method of rolling element load.

[**Example 2 – 7**] The deep groove ball bearing 6208 is subjected to radial load $F_r = 2.94\text{kN}$, steel ball number is $z = 9$, radial clearance is $G_r = 0.025\text{mm}$ and $G_r = 0.05\text{mm}$. Calculate the subjecting load on every ball. The diameter of the deep groove ball bearing 6208 is $D_w = 12.7\text{mm}$, the steel ball number is $z = 9$, the groove curvature factor is $f_i = f_e = 0.515$.

[**Resolve**] 1) From Table 2 – 7 and Equation (2 – 46) we can check out the elastic deformation constant K as:

$$K = 0.000422/D_w^{1/3} = \frac{0.000422}{12.7^{1/3}} = 0.000181$$

2) First assume that the maximum rolling element load is:

$$Q_{\max} = 1.47 \text{ kN}$$

From the equation (2-42) we can calculate:

$$\delta_{\max} = 0.000181 * 1470^{1/3} \text{ mm} = 0.0235 \text{ mm}$$

3) When the radial clearance is $G_r = 0.025 \text{ mm}$, from Equation (2-70) we can get the first approximate value of the load distribution factor T :

$$T = \frac{1}{2} \left(1 - \frac{0.025}{2 * 0.0235 + 0.025} \right) = 0.3264$$

4) From Table 2-10 we can check out the J_r value related to this T value by inserting method:

$$J_r = 0.1975$$

5) From the equation (2-65) we can calculate the first approximate $Q_{\max}$ value:

$$Q_{\max}^{(1)} = \frac{2.94 \text{ kN}}{9 * 0.1955} = 1.67 \text{ kN}$$

6) According to the first approximate $Q_{\max}$ value, again we assume:

$$Q_{\max} = 1.667 \text{ kN}$$

Then we can get:

$$\delta_{\max} = 0.0255 \text{ mm}$$

From the $\delta_{\max}$ value, we can calculate:

$$T = 0.3356$$

From Table 2-10 we can check out:

$$J_r = 0.1977$$

Again from the equation (2-65) we can get the the second approximate $Q_{\max}$ value:

$$Q_{\max}^{(2)} = 1.653 \text{ kN}$$

7) According to the second approximate $Q_{\max}$ value, again we assume:

$$Q_{\max} = 1.647 \text{ kN}$$

Then we can get:

$$\delta_{\max} = 0.0253 \text{ mm}$$

$$T = 0.3348$$

$$J_r = 0.1975$$

Again from the equation (2-65) we can get the the third approximate

$Q_{\max}$ value:

$$Q_{\max}^{(3)} = 1.654 \text{ kN}$$

Thus, taking $Q_{\max} = 1.647 \text{ kN}$ is precise enough.

$$8) \psi_1 = \frac{360^\circ}{9} = 40^\circ \quad \cos 40^\circ = 0.7660$$

From the equation (2-71) we can get:

$$Q_1 = 1.647 * \left[1 - \frac{1}{2T} (1 - \cos 40^\circ) \right]^{1.5} = 0.864 \text{ kN}$$

9) From the equation (2-72), we can calculate the load distribution angle:

$$\psi_0 = 70^\circ 45'$$

But

$$\psi_2 = 2 * \frac{360^\circ}{9} = 80^\circ > \psi_0 = 70^\circ 45'$$

So $Q_2 = Q_3 = 0$

10) Using the above same steps we can calculate the load distribution on every rolling element when the radial clearance is $G_r = 0.05 \text{ mm}$:

$$Q_{\max} = 1.833 \text{ kN}$$

$$Q_1 = 0.750 \text{ kN}$$

$$Q_2 = Q_3 = 0$$

Under $G_r = 0.025 \text{ mm}$ and $G_r = 0.050 \text{ mm}$ two conditions, the rolling element load is listed in Table 2-11.

Table 2-11 Rolling element load at different radial clearance

G_r/mm	0				
ψ	0°	$\pm 40^\circ$	$\pm 80^\circ$	$\pm 120^\circ$	$\pm 160^\circ$
Q/kN	1.43	0.96	0.103	0	0
G_r/mm	0.025				
ψ	0°	$\pm 40^\circ$	$\pm 80^\circ$	$\pm 120^\circ$	$\pm 160^\circ$
Q/kN	1.647	0.864	0	0	0
G_r/mm	0.050				
ψ	0°	$\pm 40^\circ$	$\pm 80^\circ$	$\pm 120^\circ$	$\pm 160^\circ$
Q/kN	1.833	0.750	0	0	0

Comparing the result of the example 6 and the rolling element load distribution values in Table 2-11, knowing that with the increasing of the bearing radial clearance, the maximum rolling element load is also increased.

$$\begin{aligned} G_r = 0 & \quad Q_{\max} = 1.43 \text{ kN} \\ G_r = 0.025 \text{ mm} & \quad Q_{\max} = 1.647 \text{ kN} \\ G_r = 0.050 \text{ mm} & \quad Q_{\max} = 1.833 \text{ kN} \end{aligned}$$

But the ball bearing life is inversely proportional to the 3rd power of the maximum rolling element load. So the radial clearance in bearing influences on the bearing life very much.

2.3.3 Load Distribution of Angular Contact Rolling Bearings

Angular contact ball bearing and tapered roller bearing can be subjected to both radial load and axial load at the same time, as shown in Figure 2-23. But only a single angular contact ball bearing and a single tapered roller bearing can not be subjected to pure axial load.

The load distributions in these kinds of rolling bearings are related to the load angle β . If β is smaller than some value, only some parts of ring raceway are subjected to load; if β is larger than some value, all circumferences of ring raceway are subjected to load.

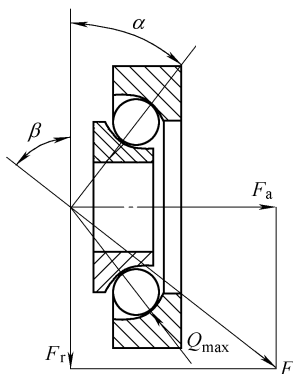


Figure 2-23 Angular contact ball bearing

Figure 2-24 is the sketch for angular contact ball bearing inner ring raceway subjected to load. The load on any rolling element can be determined by the deformation on the contact point of rolling element with ring raceway.

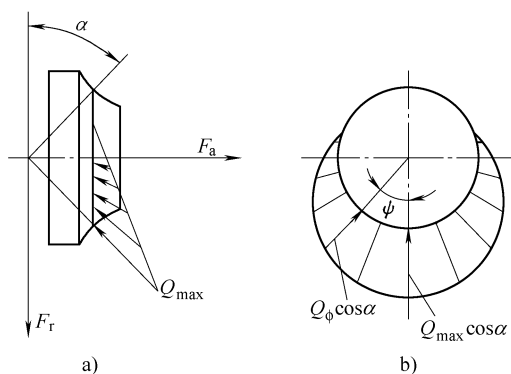


Figure 2-24 Angular contact bearing inner ring load distribution

If under radial load F_r and axial load F_a , the inner ring center translationally moves the distance δ_a and δ_r relative to the outer ring center, and the point A on the inner ring raceway that is located on the angle ψ apart from the maximum load rolling element would move to the point A_1 , as shown in Figure 2-25. Here the total deformation value is as follows:

$$\delta_{\psi} = \delta_a \sin \alpha + \delta_r \cos \alpha \cos \psi \quad (2-73)$$

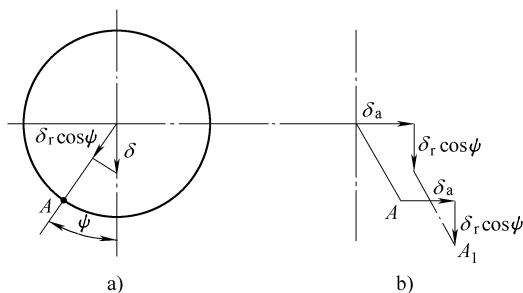


Figure 2-25 Angular contact bearing ring relative moving

$\psi = 0$ is the contact point of the maximum load rolling element with ring raceway. Here the maximum elastic deformation value is:

$$\delta_{\max} = \delta_a \sin \alpha + \delta_r \cos \alpha \quad (2-74)$$

If introducing the load distribution factor T , then:

$$T = \frac{1}{2} \left(1 + \frac{\delta_a \tan \alpha}{\delta_r} \right) \quad (2-75)$$

So, the equation (2-73) can be rewritten as:

$$\delta_i = \delta_{\max} \left[1 - \frac{1}{2T} (1 - \cos \psi) \right] \quad (2-76)$$

The load distribution T expresses the load area range in bearing, as shown in Figure 2-26.

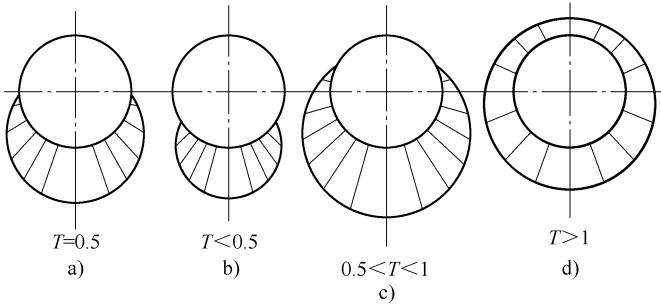


Figure 2-26 Relationship of T with angular contact bearing load distribution

$T < 1$, only part of raceway is subjecting to load;

$T = 1/2$, $\delta_a = 0$, in this condition, the inner ring and the outer ring raceways only have the relative radial displacements and only half raceway in bearing is subjecting to load;

$T \geq 1$, all of raceways are subjecting to load;

$T = \infty$, $\delta_r = 0$, in this condition, the inner ring and outer ring raceways only have the relative axial displacements, and all of rolling elements in bearing are subjecting to load;

Putting the equation (2-58) into the equation (2-76), we can get:

$$Q_\psi = Q_{\max} \left[1 - \frac{1}{2T} (1 - \cos \psi) \right]^t \quad (2-77)$$

In this equation, for ball bearing: $t = 3/2$; for roller bearing: $t = 1$.

From the equilibrium of bearing subjecting forces, we can get:

$$F_r = \sum Q_\psi \cos \alpha \cos \psi \approx z Q_{\max} \cos \alpha J_r \quad (2-78)$$

$$F_a = \sum Q_\psi \sin \alpha \approx z Q_{\max} \sin \alpha J_a \quad (2-79)$$

$$J_r = J_r(T) = \frac{1}{2\pi} \int_{-\psi_0}^{+\psi_0} \left[1 - \frac{1}{2T} (1 - \cos\psi) \right] \cos\psi d\psi \quad (2-80)$$

$$J_a = J_a(T) = \frac{1}{2\pi} \int_{-\psi_0}^{+\psi_0} \left[1 - \frac{1}{2T} (1 - \cos\psi) \right] d\psi \quad (2-81)$$

Where J_r — load distribution radial integral;

J_a — load distribution axial integral;

ψ_0 — half of load area arc.

In the equation (2-76), if $\delta_\psi = 0$, we can get:

While $T \leq 1$

$$\cos\psi_0 = 1 - 2T \quad (2-82)$$

While $T > 1$

$$\psi_0 = \pi$$

From the equations (2-78) and (2-79) we can get:

$$\frac{F_r \tan \alpha}{F_a} = \frac{J_r}{J_a} \quad (2-83)$$

$$Q_{\max} = \frac{F_r}{z \cos \alpha J_r} = \frac{F_a}{z \sin \alpha J_a} \quad (2-84)$$

The relationship of single row angular contact bearing J_r , J_a with load distribution T and $F_r \tan \alpha / F_a = \tan \alpha / \tan \beta$ are listed in Figure 2-27 and in

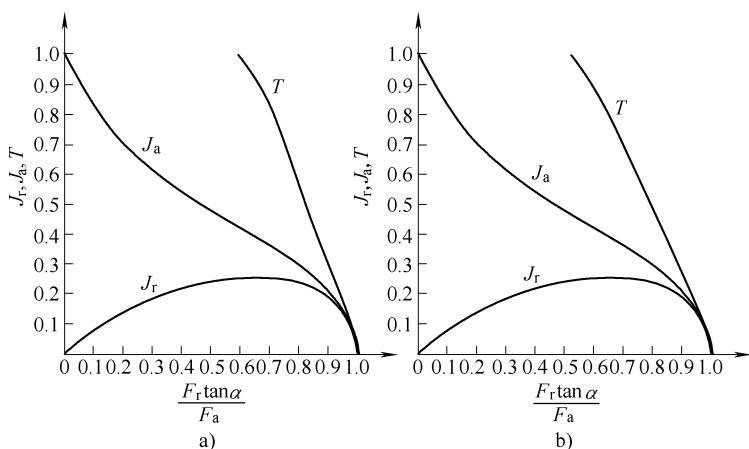


Figure 2-27 Angular contact bearing J_r , J_a

a) point contact b) line contact

Table 2 – 12. The table listed values are suitable for use in the bearing that the contact angle is not changeable under subjecting load. For the bearing that the contact angle is changeable under subjecting load, if using the actual contact angle, we also can approximately use the table listed values.

For angular contact bearing, in order to keep at least half raceway subjected to load, that is $T = 0.5$, the required axial load is:

For ball bearing: $F_a = 1.22 F_r \tan \alpha$;

For roller bearing: $F_a = 1.26 F_r \tan \alpha$.

If the axial load is lower than the above value, the load area of the rolling elements would decrease and the number of the rolling elements subjected to load would decrease. If $F_a = F_r \tan \alpha$, only one rolling element is subjecting to load.

**Table 2 – 12 Single row angular contact bearing
load distribution integral J_r , J_a**

load distribution factor	point contact			line contact		
T	$\frac{F_r \tan \alpha}{F_a}$	J_r	J_a	$\frac{F_r \tan \alpha}{F_a}$	J_r	J_a
0	1	$1/z$	$1/z$	1	$1/z$	$1/z$
0.05	0.9834	0.0828	0.0842	0.9806	0.0910	0.0928
0.1	0.9666	0.1156	0.1196	0.9613	0.1268	0.1319
0.15	0.9497	0.1397	0.1471	0.9409	0.1528	0.1624
0.2	0.9315	0.1590	0.1707	0.9216	0.1736	0.1885
0.25	0.9145	0.1753	0.1917	0.9009	0.1909	0.2119
0.3	0.8967	0.1892	0.2110	0.8805	0.2055	0.2334
0.35	0.8782	0.2012	0.2291	0.8592	0.2179	0.2536
0.4	0.8599	0.2117	0.2462	0.8380	0.2286	0.2728
0.45	0.8415	0.2209	0.2625	0.8159	0.2376	0.2912
0.5	0.8224	0.2288	0.2782	0.7939	0.2453	0.3090
0.6	0.7834	0.2416	0.3084	0.7480	0.2568	0.3433
0.7	0.7424	0.2505	0.3374	0.6999	0.2636	0.3766

(continued)

load distribution factor	point contact			line contact		
T	$\frac{F_r \tan \alpha}{F_a}$	J_r	J_a	$\frac{F_r \tan \alpha}{F_a}$	J_r	J_a
0.8	0.6996	0.2559	0.3658	0.6485	0.2658	0.4099
0.9	0.6530	0.2576	0.3945	0.5918	0.2628	0.4441
1.0	0.5999	0.2546	0.4244	0.5238	0.2523	0.4817
1.25	0.4538	0.2289	0.5044	0.3600	0.2079	0.5775
1.5	0.3551	0.2025	0.5702	0.2722	0.1755	0.6447
1.67	0.3079	0.1868	0.6067	0.2334	0.1586	0.6796
2.0	0.2440	0.1618	0.6631	0.1825	0.1334	0.7310
2.5	0.1849	0.1339	0.7240	0.1372	0.1075	0.7837
3.0	0.1487	0.1140	0.7664	0.1099	0.0900	0.8192
3.5	0.1242	0.0991	0.7977	0.0915	0.0773	0.8447
4.0	0.1066	0.0876	0.8216	0.0785	0.0678	0.8639
4.5	0.0934	0.0785	0.8405	0.0726	0.0638	0.8789
5.0	0.0831	0.0711	0.8558	0.0611	0.0544	0.8909
6.0	0.0680	0.0598	0.8790	0.0500	0.0454	0.9089
7.0	0.0576	0.0516	0.8958	0.0423	0.0390	0.9219
8.0	0.0500	0.0454	0.9085	0.0367	0.0342	0.9316
9.0	0.0441	0.0405	0.9184	0.0324	0.0304	0.9392
10	0.0394	0.0365	0.9264	0.0290	0.0274	0.9452
50	0.0076	0.0075	0.9851	0.0056	0.0055	0.9890
100	0.0037	0.0037	0.9925	0.0027	0.0027	0.9945
500	0.0007	0.0007	0.9985	0.0005	0.0005	0.9989
1000	0.0004	0.0004	0.9993	0.0003	0.0003	0.9995
∞	0	0	1	0	0	1

[**Example 2 – 8**] Single row tapered roller bearing 32315 is subjected to the radial load $F_r = 40\text{kN}$ and the axial load $F_a = 15\text{kN}$.

Calculate the maximum rolling element load $Q_{\max}$ in bearing. The 32315 bearing structure dimensions are as follows:

$$D_w = 14.989 \text{ mm}, z = 19, \alpha \approx 10^\circ 30'.$$

[**Resolve**] First we calculate:

$$\frac{F_r \tan \alpha}{F_a} = \frac{40 * 0.1853}{15} = 0.494$$

From Figure 2-27 or Table 2-12 we check out:

$$J_r = 0.244$$

From the equation (2-84) we can get:

$$Q_{\max} = \frac{F_r}{z \cos \alpha \cdot J_r} = \frac{40 \text{ kN}}{19 * 0.9833 * 0.244} = 8.77 \text{ kN}$$

2.3.4 Load Distribution in Double Row Angular Contact Rolling Bearings

Double row angular contact bearing can be subjected to any direction load, as shown in Figure 2-28. Next analysis suits for aligning bearing, such as aligning ball bearing, aligning roller bearing and double row tapered roller bearing.

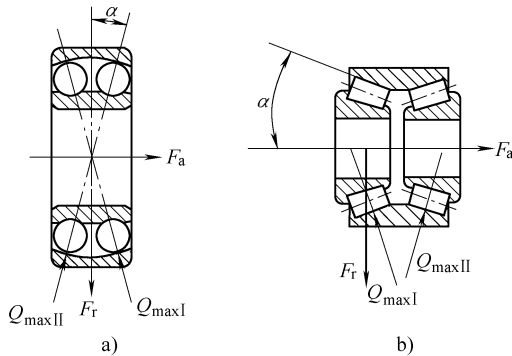


Figure 2-28 Double row angular contact bearing load distribution

a) aligning ball bearing b) double row tapered roller bearing

While subjecting to pure radial load F_r , the two row rolling elements in a bearing are subjecting to the same load. While subjecting to radial load F_r and axial load F_a at the same time, one row rolling elements are subjecting

to heavier load, but the other row rolling elements are subjecting to lighter load. Express the two rows in a bearing by corner note I , II , and assume that the row I is subjecting to heavier load, and the row II is subjecting to lighter load.

Next analysis uses the following symbols:

F_r, F_a — Radial load and axial load on bearing;

F_{rI}, F_{rII} — Radial load on the row I , II in bearing;

F_{aI}, F_{aII} — Axial load on the row I , II in bearing;

$Q_{\max I}, Q_{\max II}$ — Maximum rolling element load of the row I , II ;

T_I, T_{II} — Load distribution factor of the row I , II ;

δ_r, δ_a — Radial and axial displacement of inner ring in bearing relative to outer ring;

$\delta_{rI}, \delta_{rII}$ — Radial displacement of the inner ring I , II in bearing relative to outer ring;

$\delta_{aI}, \delta_{aII}$ — Axial displacement of the inner ring I , II in bearing relative to outer ring.

If under subjecting load, bearing inner ring displacement is δ_r and δ_a relative to outer ring, then:

$$\delta_{aI} = -\delta_{aII} = \delta_a \quad (2-85)$$

$$\delta_{rI} = \delta_{rII} = \delta_r \quad (2-86)$$

The positive sign of axial displacement means the direction along which the inner ring and outer ring are pressed for each other, and the negative sign of axial displacement means the direction along which the inner ring and outer ring are separated each other.

Because each row in a bearing can be seen as a single row angular contact radial and thrust bearing, from the equations (2-74) and (2-75) we can get:

$$\delta_{\max I} = \delta_{aI} \sin \alpha + \delta_{rI} \cos \alpha \quad (2-87)$$

$$\delta_{\max II} = \delta_{aII} \sin \alpha + \delta_{rII} \cos \alpha \quad (2-88)$$

$$T_I = \frac{1}{2} \left(1 + \frac{\delta_{aI} \tan \alpha}{\delta_{rI}} \right) \quad (2-89)$$

$$T_{II} = \frac{1}{2} \left(1 + \frac{\delta_{aII} \tan \alpha}{\delta_{rII}} \right) \quad (2-90)$$

Putting the equations (2-85) and (2-86) into the equations (2-87) to (2-90) we can get:

$$\frac{\delta_{\max \parallel}}{\delta_{\max \perp}} = \frac{T_{\parallel}}{T_{\perp}} \quad (2-91)$$

$$T_{\perp} + T_{\parallel} = 1 \quad (2-92)$$

From the equations (2-58) and (2-91) we can get:

$$\frac{Q_{\max \parallel}}{Q_{\max \perp}} = \left(\frac{\delta_{\max \parallel}}{\delta_{\max \perp}} \right)^t = \left(\frac{T_{\parallel}}{T_{\perp}} \right)^t \quad (2-93)$$

For ball bearing: $t = 3/2$; For roller bearing: $t = 1$.

From force equilibrium we can get:

$$F_r = F_{r\perp} + F_{r\parallel} = ZQ_{\max \perp} \cos \alpha J_r(T_{\perp} T_{\parallel}) \quad (2-94)$$

$$F_a = F_{a\perp} - F_{a\parallel} = ZQ_{\max \perp} \sin \alpha J_a(T_{\perp} T_{\parallel}) \quad (2-95)$$

Here, $J_r(T_{\perp} T_{\parallel})$ and $J_a(T_{\perp} T_{\parallel})$ are the radial and axial load distribution integrals of double row bearings.

$$J_r(T_{\perp} T_{\parallel}) = J_r(T_{\perp}) + \frac{Q_{\max \parallel}}{Q_{\max \perp}} J_r(T_{\parallel}) \quad (2-96)$$

$$J_a(T_{\perp} T_{\parallel}) = J_a(T_{\perp}) - \frac{Q_{\max \parallel}}{Q_{\max \perp}} J_a(T_{\parallel}) \quad (2-97)$$

In these equations, $J_r(T_{\perp})$, $J_r(T_{\parallel})$, $J_a(T_{\perp})$ and $J_a(T_{\parallel})$ are the load distribution integrals of the single row bearing, which value can be checked out from Table 2-12.

Double row bearing load distribution integral $J_r(T_{\perp} T_{\parallel})$ and $J_a(T_{\perp} T_{\parallel})$ values are listed in Table (2-13).

**Table 2-13 Double row bearing load distribution
integral $J_r(T_{\perp} T_{\parallel})$, $J_a(T_{\perp} T_{\parallel})$**

load distribution factor		point contact						line contact			
$T_{\perp}$	$T_{\parallel}$	$\frac{F_r \tan \alpha}{F_a}$	J_r	J_a	$\frac{Q_{\max \parallel}}{Q_{\max \perp}}$	$\frac{F_{r\parallel}}{F_{r\perp}}$	$\frac{F_r \tan \alpha}{F_a}$	J_r	J_a	$\frac{Q_{\max \parallel}}{Q_{\max \perp}}$	$\frac{F_{r\parallel}}{F_{r\perp}}$
0.5	0.5	∞	0.4576	0	1	1	∞	0.4906	0	1	1
0.6	0.4	2.048	0.3569	0.1743	0.5445	0.4771	2.390	0.4032	0.1687	0.6402	0.5699

(continued)

load distribu- tion factor		point contact					line contact				
T_{I}	T_{II}	$\frac{F_r \tan \alpha}{F_a}$	J_r	J_a	$\frac{Q_{\text{max II}}}{Q_{\text{max I}}}$	$\frac{F_{r \text{ II}}}{F_{r \text{ I}}}$	$\frac{F_r \tan \alpha}{F_a}$	J_r	J_a	$\frac{Q_{\text{max II}}}{Q_{\text{max I}}}$	$\frac{F_{r \text{ II}}}{F_{r \text{ I}}}$
0.7	0.3	1.092	0.3037	0.2781	0.2812	0.2124	1.210	0.3445	0.2847	0.3937	0.3069
0.8	0.2	0.8005	0.2758	0.3445	0.1249	0.0776	0.8230	0.3036	0.3689	0.2176	0.1421
0.9	0.1	0.6714	0.2619	0.3901	0.0371	0.01664	0.6340	0.2741	0.4323	0.0892	0.0430
1.0	0	0.5999	0.2546	0.4244	0	0	0.5238	0.2523	0.4817	0	0

There are three situations for bearing load:

1) $F_a = 0$. In this condition, there is only one radial load F_r , and the two row rolling elements in a bearing are subjecting to the same load.

$$F_{r I} = F_{r II} = \frac{1}{2} F_r$$

If there is no radial clearance, half raceways of the two rows in a bearing are subjecting to load. That is:

$$T_I = T_{II} = 0.5$$

For ball bearing:

$$Q_{\max I} = Q_{\max II} = 2.185 \frac{F_r}{z \cos \alpha} \quad (2-98)$$

For roller bearing:

$$Q_{\max I} = Q_{\max II} = 2.038 \frac{F_r}{z \cos \alpha} \quad (2-99)$$

2) $F_r = 0$. In this condition, there is only one axial load F_a and only one row rolling element is subjecting to load. But in this condition every rolling element is subjecting to the same load.

$$Q_I = \frac{F_a}{z \sin \alpha} \quad (2-100)$$

3) $F_a \neq 0$, $F_r \neq 0$. In this condition, the two rows in a bearing are subjecting to different loads, the row I is subjecting to heavier load and the row II is subjecting to lighter load. While:

For ball bearing: $F_a = 1.67 F_r \tan \alpha$

For roller bearing: $F_a = 1.91 F_r \tan \alpha$

Then $T_I = 1, T_{II} = 0$

Only the row (I is) subjecting to load. But the row II is not subjecting to load at all. If F_a is larger than the above value, in this time, the double row bearing functions like a single row angular contact bearing. If F_a is smaller than the above value, from the equations (2-94) and (2-95) we can get:

$$\frac{F_r \tan \alpha}{F_a} = \frac{J_r (T_I T_{II})}{J_a (T_I T_{II})} \quad (2-101)$$

$$Q_{\max I} = \frac{F_r}{z \cos \alpha J_r (T_I T_{II})} = \frac{F_a}{z \sin \alpha J_a (T_I T_{II})} \quad (2-102)$$

2.3.5 Load Distribution in Single Direction Thrust Rolling Bearings

While $\alpha = 90^\circ$ and thrust bearing is subjecting to a central axial load, every rolling element is subjecting to the same load.

$$Q = \frac{F_a}{z} \quad (2-103)$$

Single direction thrust bearing can be subjecting to an eccentric load, as shown in Figure 2-29. An eccentric load can be resolved into one central axial load F_a and one moment through the plane of the bearing central axis and the load acting line, that is:

$$M = e F_a \quad (2-104)$$

Under the central axial load F_a acting, the bearing shaft washer axially moves δ_a relative to the housing washer; under the moment M acting, the two washers relative to each other turn an angle θ . So in this condition, every rolling element is subjecting to different load.

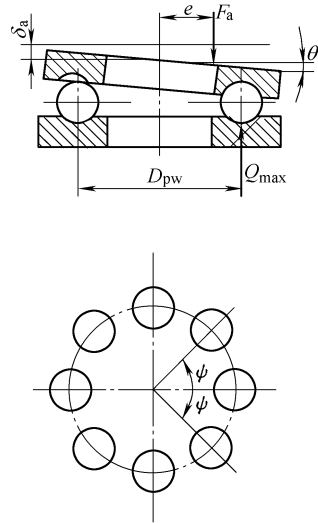


Figure 2-29 Single direction thrust bearing load distribution

$$\delta_{\max} = \delta_a + \frac{1}{2} D_{pw} \theta \quad (2-105)$$

$$\delta_{\psi} = \delta_a + \frac{1}{2} D_{pw} \theta \cos \psi \quad (2-106)$$

If introducing load distribution factor T

$$T = \frac{1}{2} \left(1 + \frac{2\delta_a}{D_{pw} \theta} \right) \quad (2-107)$$

Then the equation (2-106) can be rewritten as:

$$\delta_{\psi} = \delta_{\max} \left[1 - \frac{1}{2T} (1 - \cos \psi) \right] \quad (2-108)$$

Putting equation (2-58) in, we can get:

$$Q_{\psi} = Q_{\max} \left[1 - \frac{1}{2T} (1 - \cos \psi) \right]^t \quad (2-109)$$

For ball bearing: $t = 3/2$; For roller bearing: $t = 1$.

From force equilibrium we can get:

$$F_a = \sum Q_{\psi} \quad (2-110)$$

$$M = e F_a = \sum \frac{1}{2} Q_{\psi} D_{pw} \cos \psi \quad (2-111)$$

Putting equation (2-109) in and introducing load distribution integral J_a, J_M , we can get:

$$F_a = z Q_{\max} J_a \quad (2-112)$$

$$M = \frac{1}{2} z Q_{\max} D_{pw} J_M \quad (2-113)$$

Here
$$J_a = \frac{1}{2\pi} \int_{-\psi_0}^{+\psi_0} \left[1 - \frac{1}{2T} (1 - \cos \psi) \right]^t d\psi \quad (2-114)$$

$$J_M = \frac{1}{2\pi} \int_{-\psi_0}^{+\psi_0} \left[1 - \frac{1}{2T} (1 - \cos \psi) \right]^t \cos \psi d\psi \quad (2-115)$$

From the equations (2-112) and (2-113) we can get:

$$\frac{2e}{D_{pw}} = \frac{J_M}{J_a} \quad (2-116)$$

Table 2-14 lists the relationship of $2e/D_{pw}$ and J_M, J_a . The values of J_M, J_a also can be checked out from Figure 2-30.

Table 2 – 14 Single direction thrust bearing ($\alpha = 90^\circ$)
load distribution integral J_M , J_a .

Load distribution factor T	Ball bearing			Roller bearing		
	$\frac{2e}{D_{pw}}$	J_M	J_a	$\frac{2e}{D_{pw}}$	J_M	J_a
0	1	1/z	1/z	1	1/z	1/z
0.05	0.9834	0.0828	0.0842	0.9806	0.0910	0.0928
0.1	0.9666	0.1156	0.1196	0.9613	0.1268	0.1319
0.15	0.9497	0.1397	0.1471	0.9409	0.1528	0.1624
0.2	0.9315	0.1590	0.1707	0.9216	0.1736	0.1885
0.25	0.9145	0.1753	0.1917	0.9009	0.1909	0.2119
0.3	0.8967	0.1892	0.2110	0.8805	0.2055	0.2334
0.35	0.8782	0.2012	0.2291	0.8592	0.2179	0.2536
0.4	0.8599	0.2117	0.2462	0.8380	0.2286	0.2728
0.45	0.8415	0.2209	0.2625	0.8519	0.2376	0.2912
0.5	0.8224	0.2288	0.2782	0.7939	0.2453	0.3090
0.6	0.7834	0.2416	0.3084	0.7480	0.2568	0.3433
0.7	0.7424	0.2505	0.3374	0.6999	0.2636	0.3766
0.8	0.6996	0.2559	0.3658	0.6485	0.2658	0.4099
0.9	0.6530	0.2576	0.3945	0.5918	0.2628	0.4441
1.0	0.5999	0.2546	0.4244	0.5238	0.2523	0.4817
1.25	0.4538	0.2289	0.5044	0.3600	0.2079	0.5775
1.5	0.3551	0.2025	0.5702	0.2722	0.1755	0.6447
1.67	0.3079	0.1868	0.6067	0.2334	0.1586	0.6796
2.0	0.2440	0.1618	0.6631	0.1825	0.1334	0.7310
2.5	0.1849	0.1339	0.7240	0.1372	0.1075	0.7837
3.0	0.1487	0.1140	0.7664	0.1099	0.0900	0.8192
3.5	0.1242	0.0991	0.7977	0.0915	0.0773	0.8447
4.0	0.1066	0.0876	0.8216	0.0785	0.0678	0.8639
4.5	0.0934	0.0785	0.8405	0.0726	0.0638	0.8789
5.0	0.0831	0.0711	0.8558	0.0611	0.0544	0.8909

(continued)

Load distribution factor T	Ball bearing			Roller bearing		
	$\frac{2e}{D_{pw}}$	J_M	J_a	$\frac{2e}{D_{pw}}$	J_M	J_a
6.0	0.0680	0.0598	0.8790	0.0500	0.0454	0.9089
7.0	0.0576	0.0516	0.8958	0.0423	0.0390	0.9219
8.0	0.0500	0.0454	0.9085	0.0367	0.0342	0.9316
9.0	0.0441	0.0405	0.9184	0.0324	0.0304	0.9392
10	0.0394	0.0365	0.9264	0.0290	0.0274	0.9452
50	0.0076	0.0075	0.9851	0.0056	0.0055	0.9890
100	0.0037	0.0037	0.9925	0.0027	0.0027	0.9945
500	0.0007	0.0007	0.9985	0.0005	0.0005	0.9989
1000	0.0004	0.0004	0.9993	0.0003	0.0003	0.9995
∞	0	0	1	0	0	1

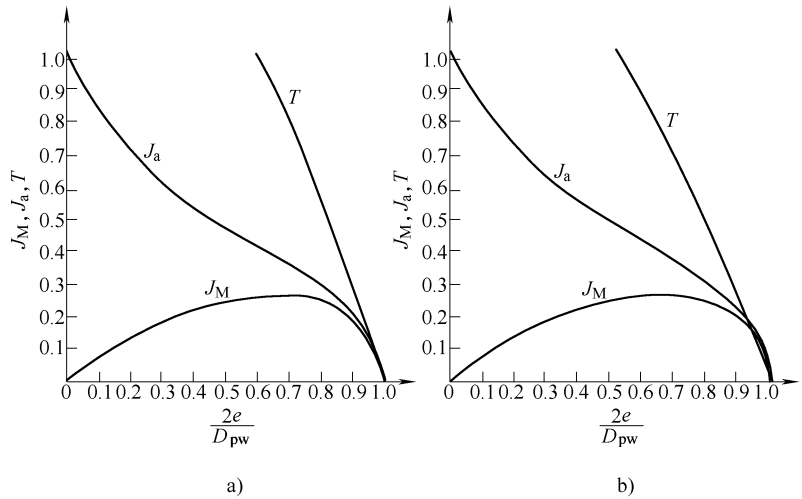


Figure 2 – 30 Single direction thrust bearing ($\alpha = 90^\circ$)
load distribution integral J_M , J_a .
a) ball bearing b) roller bearing

From the values of Table 2 – 14 we can get:

while $2e/D_{pM} = 0.8224$,

$$T = 0.5 \quad J_a = 0.2782$$

In this condition, there are only half raceways subjecting to load, from the equation (2-112) we can get:

$$Q_{\max} = 3.6 \frac{F_a}{z} \quad (2-117)$$

While $2e/D_{pM} = 1$, there are only one rolling element subjecting to load.

If $2e > D_{pM}$, the bearing rings would be inclined and separated. Therefore, single direction thrust bearing only can be subjecting to an eccentric load of $e \leq 1/2 D_{pM}$.

2.3.6 Load Distribution in Double Direction Thrust Rolling Bearings

While a double direction thrust bearing subjects to a central axial load, there are only one row rolling elements subjecting to the load. In this condition, we can calculate the load distribution according to single direction thrust bearing.

Double direction thrust bearing can subject to an arbitrary eccentric load, as shown in Figure 2-31.

In double direction thrust bearing:

$$\delta_{aI} = -\delta_{aII} = \delta_a \quad (2-118)$$

$$\theta_I = \theta_{II} = \theta \quad (2-119)$$

Here, δ_{aI} and δ_{aII} are the axial displacement of row I and II respectively, and θ_I and θ_{II} are the incline angle of row I and II respectively.

From the equation (2-107) we can get:

$$T_I = \frac{1}{2} \left(1 + \frac{2\delta_a}{D_{pw}\theta} \right) \quad (2-120)$$

$$T_{II} = \frac{1}{2} \left(1 - \frac{2\delta_a}{D_{pw}\theta} \right) \quad (2-121)$$

From the equations (2-120) and (2-121) we can get:

$$T_I + T_{II} = 1$$

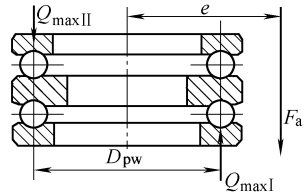


Figure 2-31 Double direction thrust bearing load distribution

From the equation (2-105) we can get:

$$\delta_{\max I} = \delta_a + \frac{1}{2} D_{pw} \theta = \frac{1}{2} D_{pw} \theta T_I \quad (2-122)$$

$$\delta_{\max II} = -\delta_a + \frac{1}{2} D_{pw} \theta = \frac{1}{2} D_{pw} \theta T_{II} \quad (2-123)$$

so
$$\frac{\delta_{\max II}}{\delta_{\max I}} = \frac{T_{II}}{T_I} \quad (2-124)$$

But
$$\frac{Q_{\max II}}{Q_{\max I}} = \left(\frac{\delta_{\max II}}{\delta_{\max I}} \right)^t = \left(\frac{T_{II}}{T_I} \right)^t \quad (2-125)$$

For ball bearing: $t = 3/2$; For roller bearing: $t = 1$.

From force equilibrium we can get:

$$F_a = F_{aI} - F_{aII} = z Q_{\max I} J_a (T_I T_{II}) \quad (2-126)$$

$$M = M_I + M_{II} = \frac{1}{2} z Q_{\max I} D_{pw} J_M (T_I T_{II}) \quad (2-127)$$

Here
$$J_a (T_I T_{II}) = J_a (T_I) - \frac{Q_{\max II}}{Q_{\max I}} J_a (T_{II}) \quad (2-128)$$

$$J_M (T_I T_{II}) = J_M (T_I) + \frac{Q_{\max II}}{Q_{\max I}} J_M (T_{II}) \quad (2-129)$$

Because
$$M = e F_a$$

From the equations (2-126) and (2-127) we can get:

$$\frac{2e}{D_{pw}} = \frac{J_M (T_I T_{II})}{J_a (T_I T_{II})} \quad (2-130)$$

The relationship of $J_a (T_I T_{II})$, $J_M (T_I T_{II})$, $\frac{2e}{D_{pw}}$, T_I and T_{II} are listed in Table 2-15, where we can see that double direction thrust bearing can subject to an arbitrary eccentric load.

Table 2-15 Double direction thrust bearing load distribution $J_M (T_I T_{II})$ and $J_a (T_I T_{II})$

Load distribution factor		Ball bearing					Roller bearing				
T_I	T_{II}	$\frac{2e}{D_m}$	J_M	J_a	$\frac{Q_{\max II}}{Q_{\max I}}$	$\frac{M_{II}}{M_I}$	$\frac{2e}{D_m}$	J_M	J_a	$\frac{Q_{\max II}}{Q_{\max I}}$	$\frac{M_{II}}{M_I}$
0.5	0.5	∞	0.4577	0	1	1	∞	0.4906	0	1	1

(continued)

Load distribution factor		Ball bearing					Roller bearing				
T_I	T_{II}	$\frac{2e}{D_m}$	J_M	J_a	$\frac{Q_{\max II}}{Q_{\max I}}$	$\frac{M_{II}}{M_I}$	$\frac{2e}{D_m}$	J_M	J_a	$\frac{Q_{\max II}}{Q_{\max I}}$	$\frac{M_{II}}{M_I}$
0.6	0.4	2.046	0.3568	0.1744	0.544	0.477	2.389	0.4031	0.1687	0.640	0.570
0.7	0.3	1.092	0.3036	0.2782	0.281	0.212	1.210	0.3445	0.2847	0.394	0.306
0.8	0.2	0.8005	0.2758	0.3445	0.125	0.078	0.8232	0.3036	0.3688	0.218	0.142
0.9	0.1	0.6713	0.2618	0.3900	0.037	0.017	0.6342	0.2741	0.4321	0.089	0.043
1.0	0	0.6000	0.2546	0.4244	0	0	0.5238	0.2523	0.4817	0	0

While for ball bearing $e \leq 0.3 D_{pw}$ and for roller bearing $e \leq 0.2619 D_{pw}$, there are only one row rolling elements subjecting to load. In this condition, the maximum rolling element load can be calculated according to Table 2 – 14. If the eccentric distance is larger than the above value, the other row rolling elements would also subject to load.

Double direction thrust bearing can subject to pure moment. In this condition, $e = \infty$, and from Table 2 – 15 we can get:

For Double direction thrust ball bearing:

$$J_M (T_I T_{II}) = 0.4577$$

$$Q_{\max I} = Q_{\max II} = \frac{2M}{zD_{pw}J_M (T_I T_{II})} = \frac{4.37M}{zD_{pw}} \quad (2-131)$$

For Double direction thrust roller bearing:

$$J_M (T_I T_{II}) = 0.4906$$

$$Q_{\max I} = Q_{\max II} = \frac{2M}{zD_{pw}J_M (T_I T_{II})} = \frac{4.08M}{zD_{pw}} \quad (2-132)$$

2.4 Kinematics of Rolling Bearings

2.4.1 Simple Kinematics Relations in Rolling Bearings

Rolling bearings are used to support load while permitting rotational motion of a shaft. In order to analyze the bearing rotating situation, the relative kinematic relationship of every component has to be known. Figure

2-32 is an angular contact bearing, which contact angle is α .

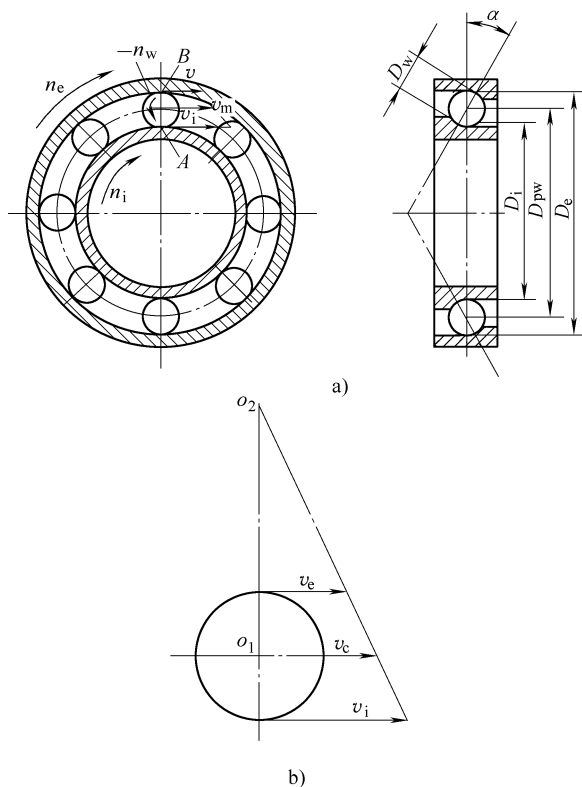


Figure 2-32 Angular contact bearing kinematic relation

a) kinematic relation b) ball velocity

Assume that the clockwise rotating direction of every component is positive, and the counterclockwise rotating direction is negative. The symbol meanings are as follows:

- n_i — Inner ring rotational speed (r/min);
- n_e — Outer ring rotational speed (r/min);
- n_c — Rolling element rotational speed about bearing axis, also called as rolling element common revolution speed. Because the cage rotation is brought by the rolling element rotation, it is also

called as cage rotational speed (r/min) ;

n_w — Rolling element rotational speed about its own axis, also called as rolling element self rotational speed (r/min) ;

n_{ci} — Cage rotational speed relative to inner ring (r/min) ;

n_{ec} — Outer ring rotational speed relative to cage (r/min) ;

v_i — Circumference velocity of inner ring raceway contact point (mm/s) ;

v_e — Circumference velocity of outer ring raceway contact point (mm/s) ;

v_c — Circumference velocity of rolling element center (mm/s) ;

ω_i — Inner ring angular speed (1/s) ;

ω_e — outer ring angular speed (1/s) ;

ω_c — Rolling element common revolution angular speed, also as cage angular speed (1/s) ;

ω_w — Rolling element self revolution angular speed (1/s) ;

D_i — Inner ring contact point diameter (mm) ;

D_e — Outer ring contact point diameter (mm) ;

D_{pw} — Rolling element center circle diameter, that is the bearing average diameter (mm) ;

D_w — Rolling element diameter (mm) ;

α — Contact angle [rad or ($^\circ$)] .

Relationship between rotational speed n and angular speed ω is as follows:

$$\omega = \frac{2\pi n}{60}$$

Knowing from Figure 2 - 32: the circumference velocity of the outer ring contact point B is as follows:

$$v_e = \frac{1}{2} D_e \omega_e = \frac{\pi}{60} n_e D_e$$

The circumference velocity of the inner ring contact point A is as follows:

$$v_i = \frac{1}{2} D_i \omega_i = \frac{\pi}{60} n_i D_i$$

If introducing:
$$\gamma = \frac{D_w \cos \alpha}{D_{pw}} \quad (2-17)$$

Then:
$$D_e = D_{pw} + D_w \cos \alpha = D_{pw} (1 + \gamma) \quad (2-133)$$

$$D_i = D_{pw} - D_w \cos \alpha = D_{pw} (1 - \gamma) \quad (2-134)$$

According to the condition of pure rolling motion, the linear velocity on the point A and the point B of the rolling element should be the same value with the point A on the inner ring raceway contact point and the point B on the outer ring raceway contact point, and v_i and v_e also is the linear velocity on the corresponding point of the rolling element.

(1) Cage rotational speed n_c

Knowing from Figure 2-32 b, the instantaneous rotating center o_2 can be found from the instantaneous velocity of the point A and the point B on the rolling element. From the velocity figure we can get:

$$v_c = \frac{v_i + v_e}{2} = \frac{\pi}{120} D_{pw} [n_i (1 - \gamma) + n_e (1 + \gamma)]$$

But as the point o_1 on cage, its linear velocity should be:

$$v_c = \frac{\pi D_{pw}}{60} n_c$$

From the above two equations we can get:

$$n_c = \frac{1}{2} [n_i (1 - \gamma) + n_e (1 + \gamma)] \quad (2-135)$$

(2) Cage rotational speed relative to inner ring n_{ci}

If adding a $(-n_i)$ rotational speed to each element of making relative moving, the relative moving relationship keeps not changing. In this relative moving relationship, the cage rotational speed relating to the inner ring is:

$$n_{ci} = n_c - n_i = \frac{1}{2} (n_c - n_i) (1 + \gamma) \quad (2-136)$$

For the same reason, we can get the inner ring rotational speed relative to cage:

$$n_{ic} = n_i - n_c = -n_{ci}$$

(3) Outer ring rotational speed n_{ec} relative to cage

This rotational speed is equivalent to the outer ring rotational speed while the cage rotational speed is zero:

$$n_{ec} = n_e - n_c = \frac{1}{2} (n_e - n_i) (1 - \gamma) \quad (2-137)$$

For the same reason, we can get the cage rotational speed relative to outer ring:

$$n_{ce} = n_c - n_e = -n_{ec}$$

(4) Rolling element self rotational speed n_w

This rotational speed is equivalent to the rolling element rotational speed about its own center axis while the cage rotational speed is zero, which can be obtained from that the linear velocity of the contact point of two contacting bodies is equal.

In making relative moving, the linear velocity of the point A on the rolling element should be:

$$V_{wA} = -\frac{2\pi n_w}{60} * \frac{D_w}{2}$$

The linear velocity of point A on the inner ring should be:

$$v_{iA} = \frac{2\pi (n_i - n_c)}{60} * \frac{D_i}{2}$$

Because $v_{wA} = v_{iA}$

$$\text{So } n_w = \frac{D_{pw}}{2D_w} (n_e - n_i) (1 + \gamma) (1 - \gamma) \quad (2-138)$$

In actual application, it is very scarce to have inner ring and outer ring rotating at the same time. Usually only one ring is rotating, the other ring is static. If assume that the inner ring is rotating, the outer ring is static, and the shaft rotational speed is n , then

$$n_i = n \quad n_e = 0$$

From the equations (2-135) and (2-138) we can get:

$$n_c = \frac{1}{2} n (1 - \gamma) \quad (2-139)$$

$$n_{ci} = -\frac{1}{2} n (1 + \gamma) \quad (2-140)$$

$$n_{ec} = -\frac{1}{2} n (1 - \gamma) \quad (2-141)$$

$$n_w = -\frac{D_{pw}}{2D_w} n (1 + \gamma) (1 - \gamma) \quad (2-142)$$

In thrust bearing ($\alpha = 90^\circ$, $\gamma = 0$), if the shaft washer is rotating, the housing washer is static, and the shaft rotational speed is n , then

$$\begin{aligned} n_i &= n & n_e &= 0 \\ n_c &= \frac{1}{2}n \end{aligned} \quad (2-143)$$

$$n_{ci} = -\frac{1}{2}n \quad (2-144)$$

$$n_{ec} = -\frac{1}{2}n \quad (2-145)$$

$$n_w = -\frac{D_{pw}}{2D_w}n \quad (2-146)$$

Under the general bearing rotational speed, the measured results in reference [27] are conformed with the above calculation results. But if in high speed bearing condition, it also should be considered that the inertia toque and friction toque would influence on every element motion.

2.4.2 Rolling Contact Number

In order to make decision of bearing life and carrying load capacity, it is required to know how many rolling elements would pass one point of a ring during this ring rotation for one circle.

Assume that the inner ring is rotating, the outer ring is in static, and the shaft rotational speed is n , that is:

$$n_i = n \quad n_e = 0$$

When the cage is rotating one circle relative to the inner ring or the outer ring, there have been z rolling elements passing one point of the inner ring or the outer ring. So, when the inner ring is rotating one circle, the rolling element numbers passing one point of the inner ring or outer ring are:

$$u_i = \frac{zn_{ci}}{n} = \frac{z}{2} (1 + \gamma) \quad (2-147)$$

$$u_e = \frac{zn_{ec}}{n} = \frac{z}{2} (1 - \gamma) \quad (2-148)$$

Here u_i — Rolling contact number of one point of inner ring raceway;
 u_e — Rolling contact number of one point of outer ring raceway.

In the calculation, u_i and u_e are always taken positive.

2.4.3 Stress Cycling Number

Stress cycling number designates how many times one point on ring raceway subjects to stress after bearing rotating certain rotations. So, stress cycling number is related to the number of rolling elements subjecting to load in loading area. Note that we don't make confusion of stress cycling number with rolling contact number.

Assume that the inner ring is rotating, the outer ring is in static, the bearing is subjecting to fixed radial load and axial load, and the central angle that the bearing loading area is facing to is $2\psi_0$ (see Figure 2-21).

L designates the bearing rotation expressed by million rotations. N_i , N_e designate stress cycling numbers of one point on inner ring and outer ring (respectively) after bearing rotating L million rotations. Then

$$N_i = u_i L \frac{2\psi_0}{360^\circ} \quad (2-149)$$

$$N_e = u_e L \quad (2-150)$$

Here u_i , u_e are one the rolling contact number of one point on the inner ring and outer ring respectively, and can be calculated from the equations (2-147) and (2-148).

2.5 Contact Angle and Axial Load Capacity in Radial and Angular Contact Ball Bearings

Because there is a radial clearance in a radial ball bearing, when its inner ring makes axial moving relative to its outer ring, there will be forming an initial contact angle α_0 . While the axial load F_a is applied, the actual contact angle will become larger. After an axial load is applied to an angular contact ball bearing, its contact angle also will become larger. So, the contact angle change should be considered when determining the axial load carrying capacity for not cutting off the ellipse area.

2.5.1 Relationship of Radial Ball Bearing Radial Clearance with Initial Contact Angle

Figure 2-33a shows a radial ball bearing with a radial clearance G_r . Figure 2-33b shows the forming of an initial contact angle α_0 of a steel ball contacting with inner ring and outer ring raceways when the inner ring makes an axial moving $G_a/2$ relative to the outer ring.

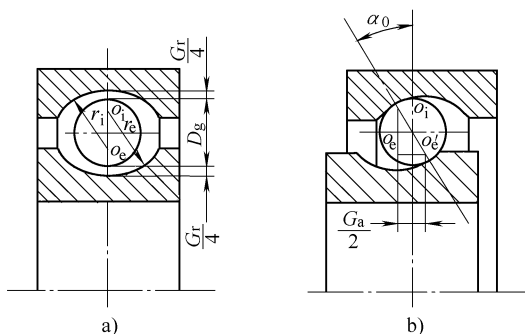


Figure 2-33 Radial clearance with contact angle

a) Radial clearance b) Axial clearance

In Figure 2-33:

o_i — Inner ring raceway curvature center;

o_e — Outer ring raceway curvature center;

o'_e — Outer ring raceway curvature center When the inner ring makes an axial moving G_a/z relative to the outer ring;

G_r — Bearing radial clearance (mm);

G_a — Bearing axial clearance (mm).

From $\triangle o_i o_e o'_e$ geometry relation ship we can get:

$$\cos \alpha_0 = 1 - \frac{G_r}{2qD_w} \quad (2-151)$$

Here

$$q = f_i + f_e - 1 \quad (2-152)$$

$$f_i = \frac{r_i}{D_w} \quad f_e = \frac{r_e}{D_w}$$

D_w — steel ball diameter (mm).

$$\delta_n = KQ^{2/3}$$

Here K is a constant. If omitting the influence of the inner ring raceway curvature in the normal plane at the contact point, and having the same raceway curvature radius for the inner and outer ring in the axial plane, then:

$$K = k \frac{1}{D_w^{1/3}}$$

Here, the k value can be checked out from Table 2-7 or Figure 2-16.

Q is the rolling element load, under a central axial load F_a applied

$$Q = \frac{F_a}{z \sin \alpha}$$

From Figure 2-34 geometry relationship we can get:

$$o_i o_e \cos \alpha_0 = o'_i o_e \cos \alpha$$

Putting the above K and Q value into the above equation, we can get:

$$\left(\frac{\cos \alpha_0}{\cos \alpha} - 1 \right) \sin^{2/3} \alpha = \frac{k}{q} \left(\frac{F_a}{z D_w^2} \right)^{2/3} \quad (2-154)$$

For certain given angular contact ball bearing, if knowing F_a , then we can get the actual contact angle α value from the equation (2-154).

For radial ball bearing with radial clearance $G_r = 0$, its initial contact angle $\alpha_0 = 0$, and the equation (2-154) can be written as:

$$\left(\frac{1}{\cos \alpha} - 1 \right) \sin^{2/3} \alpha = \frac{K}{q} \left(\frac{F_a}{z D_w^2} \right)^{2/3}$$

Through trigonometric function transformation:

$$2 \sin^{2/3} \alpha \left(\frac{1}{\cos \alpha} - 1 \right) = 2 \tan^{2/3} \alpha (1 + \tan^2 \alpha)^{1/6} \left(1 - \frac{1}{\sqrt{1 + \tan^2 \alpha}} \right)$$

Using series expansion equation:

$$(1 + \tan^2 \alpha)^{1/6} = 1 + \frac{1}{6} \tan^2 \alpha - \frac{5}{72} \tan^4 \alpha + \dots$$

$$\frac{1}{\sqrt{1 + \tan^2 \alpha}} = 1 - \frac{1}{2} \tan^2 \alpha + \frac{3}{8} \tan^4 \alpha + \dots$$

Omitting high order micro-value, we can get:

$$\tan \alpha = \left(\frac{2K}{q} \right)^{3/8} \left(\frac{F_a}{zD_w^2} \right)^{1/4} \quad (2-155)$$

While subjecting to radial load and axial load at the same time, in angular contact ball bearing, the contact angle changes with rolling element position.

2.5.3 Safety Contact Angle and Axial Load Capacity for Radial Ball Bearings and Angular Contact Ball Bearings

Deep groove ball bearings and angular contact ball bearings have been extensively used. These type bearings can subject to combined loads of radial load and axial load at the same time, and also can subject to pure axial load. Under a combined load or an axial load, there is forming a contact angle between the ball and the inner ring raceway or outer ring raceway. If the axial load is exceedingly large, a part of the contact ellipse would be cut off by the ring rib. So there would exist a stress concentration in the vicinity of contacting area between the ball and the ring ribs, and the bearing would get fatigue failure very quickly.

For a given radial ball bearing or an angular contact ball bearing, it is right to decide its axial direction carrying capacity by using this condition. The contact angle where the contact ellipse is just arriving at the position of ring rib and no stress concentration is generated, is the safety contact angle. In Figure 2-35, α is the safety contact angle where the contact ellipse is just arriving at the position of ring rib.

The angle θ can be approximately calculated by the following equation:

For outer ring

$$\cos \theta_e = 1 - \frac{D_e - D_{el}}{D_w} \quad (2-156)$$

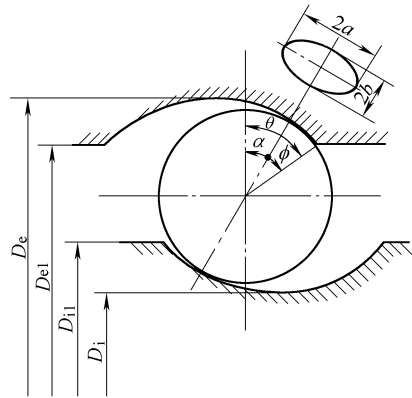


Figure 2-35 Safety contact angle

$$\text{For inner ring} \quad \cos\theta_i = 1 - \frac{D_{il} - D_i}{D_w} \quad (2-157)$$

Angle φ is the angle formed by the contact ellipse semi-major axis and ball center. After elastic deformation, the ball radius has being changed very little. So it can be approximately calculated by the following equation:

$$\sin\phi = \frac{2a}{D_w} \quad (2-158)$$

Here a — contact ellipse semi-major axis (mm);

D_w — ball diameter (mm).

In order to guarantee the bearing normal rotating, bearing actual contact angle should meet the following equation:

$$\alpha \leq \theta - \phi \quad (2-159)$$

Knowing from the equation (2-30):

$$a = C_a \sqrt[3]{\frac{Q}{\Sigma\rho}}$$

Here, C_a and $\Sigma\rho$ are related to contact condition, mainly related to osculation. Contact ellipse semi-major axis gets larger with the osculation getting larger. Therefore while the osculation gets larger, the permissible safety contact angle of the radial ball bearings and angular contact ball bearings gets smaller, that is the bearing axial direction carrying capacity becomes smaller.

For a given ball bearing, its axial direction carrying capacity is related to rolling element loaded Q , which is related to contact angle. The contact angle is related to the applied axial load F_a on bearing and bearing radial clearance. So it is very complicated to precisely calculate the bearing axial direction carrying capacity. We can only approximately decide it under pure axial load.

Under pure axial load, from the equations (2-158), (2-159) and (2-30) we can get:

$$\sin(\theta - \alpha) = \frac{2C_a}{D_w} \left(\frac{F_a}{z \sin\alpha \Sigma\rho} \right)^{1/3}$$

From conversion, we can get:

$$F_a = \left(\frac{D_w \sin(\theta - \alpha)}{2C_a} \right)^3 z \sin \alpha \Sigma \rho \quad (2-160)$$

If α is the safety contact angle in this equation, then F_a is the bearing axial direction carrying capacity.

From the equation (2-154) we can get:

$$F_a = \left[\frac{q}{k} \left(\frac{\cos \alpha_0}{\cos \alpha} - 1 \right) \right]^{1.5} z D_w^2 \sin \alpha \quad (2-161)$$

From the equations (2-160) and (2-161) we can get:

$$\sin(\theta - \alpha) = \frac{2C_a \left[\frac{q}{k} \left(\frac{\cos \alpha_0}{\cos \alpha} - 1 \right) \right]^{1/2}}{(D_w \Sigma \rho)^{1/3}} \quad (2-162)$$

From the equation (2-162) we can approximately calculate the safety contact angle. Putting the calculated safety contact angle α into the equation (2-160), we can get the maximum bearing axial direction carrying capacity F_a value.

Summarizing the above mentioned, the steps to decide the axial direction carrying capacity for radial ball bearing and angular contact ball bearing are as follows:

1) According to bearing geometry dimensions, decide θ value from the equations (2-156) and (2-157).

2) According to radial clearance G_r , decide initial contact angle α_0 from the equation (2-151).

3) Decide safety contact angle from the equation (2-162).

4) Decide bearing axial direction carrying capacity F_a value from the equation (2-160).

[**Example 2-9**] Decide axial direction carrying capacity for 6008 radial ball bearing. The 6008 bearing structure and dimensions are listed in Figure 2-36.

Radial clearance $G_r = 0.051 \text{ mm}$	$z = 12$
$D_w = 7.938 \text{ mm}$	$r_e = r_i = 4.09 \text{ mm}$
$D_e = 61.938 \text{ mm}$	$D_{eI} = 58.8 \text{ mm}$
$D_i = 46.062 \text{ mm}$	$D_{iI} = 49.2 \text{ mm}$

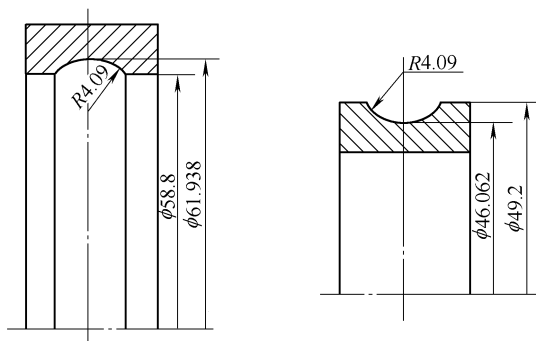


Figure 2-36 6008 bearing structure and dimensions

[**Resolve**] (1) Decide θ_e , θ_i

From the equation (2-156) we can get:

$$\cos\theta_e = 1 - \frac{D_e - D_{e1}}{D_w} = 1 - \frac{61.938 - 58.8}{7.938} = 0.6047$$

From the equation (2-157) we can get:

$$\cos\theta_i = 1 - \frac{D_{i1} - D_i}{D_w} = 0.6047$$

$$\theta_e = \theta_i = 52^\circ 47'$$

(2) Decide α_0

From the equation (2-151) we can get:

$$\cos\alpha_0 = 1 - \frac{G_r}{2qD_w} = 1 - \frac{0.051}{2 * 0.030 * 7.938} = 0.8929$$

$$\alpha_0 = 26^\circ 44'$$

(3) From the equation (2-18) we can get:

$$\Sigma\rho_i = 0.297$$

$$\Sigma\rho_e = 0.230$$

From the equation (2-20) we can get:

$$F(\rho_i) = 0.951$$

$$F(\rho_e) = 0.936$$

From Table 2-4 we can check out:

$$C_{ai} = 0.09794 \quad C_{ae} = 0.08803$$

From the equation (2-152) we can get:

$$f_i = f_e = 0.515 \quad q = 0.030$$

From Table 2-7 we can get:

$$k = 0.000422$$

On the contact point of the ball with the inner ring raceway, from the equation (2-162) we can get:

$$\sin(\theta - \alpha) = 0.3924 * \left(\frac{0.8929}{\cos \alpha} - 1 \right)^{1/2}$$

On the contact point of the ball with the outer ring raceway:

$$\sin(\theta - \alpha) = 0.4594 * \left(\frac{0.8929}{\cos \alpha} - 1 \right)^{1/2}$$

If assuming $\alpha = 34^\circ$, putting it into the above equation can get:

$$\theta - \alpha = 19^\circ 49' \quad \text{then } \alpha = 32^\circ 58'$$

If assuming $\alpha = 33^\circ 30'$, putting it into the above equation can get:

$$\theta - \alpha = 19^\circ 14' \quad \text{then } \alpha = 33^\circ 33'$$

If assuming $\alpha = 33^\circ 33'$, putting it into the above equation can get:

$$\theta - \alpha = 19^\circ 18' \quad \text{then } \alpha = 33^\circ 29'$$

Therefore, taking $\alpha = 33^\circ 30'$ as the safety contact angle at the contact point of the ball with the inner ring raceway is precise enough.

Using the same calculation method also can get that the safety contact angle at the contact point of the ball contacting with the outer ring raceway is $\alpha = 33^\circ 40'$.

Putting $\alpha = 33^\circ 30'$ into the equation (2-160), we can get the 6008 radial ball bearing axial direction load carrying capacity:

$$F_a = 4725 \text{ N}$$

2.6 Lundberg-Palmgren Theory on Rolling Bearing Dynamic Rating and Life

2.6.1 Main Failure Modes of Rolling Bearings

A rolling bearing can not give unlimited service forever. There are many factors that could cause bearing early damage, such as not properly mounting, not properly lubrication, water moisture and dust intruding, rust, electric rust and so on. If mounting, lubrication and maintainance are in

normal condition, after certain period rotating, bearing can appear fatigue failure and wear that can cause abnormal working.

Main failure modes of rolling bearings are as follows:

(1) Fatigue spalling

In rolling bearing, the contact surfaces (raceway surface and rolling element surface) are subjected to load and also making relative motion. Because of the stresses repeatedly acting on these surfaces, at first, fatigue cracking commences at the strength weak points below the surface in certain depth. This crack then propagates to the surface, eventually forming metal flakes falling from the surfaces of the raceways and rolling elements in the load-carrying surface to let bearing not normally working. This phenomenon is called as fatigue spalling. It is unavoidable that rolling bearing get fatigue spalling at last. In fact, under normally mounted, lubricated, and sealed, most of rolling bearings suffer fatigue failure. So, usually rolling bearing life is a designation for bearing fatigue life.

(2) Plastic deformation (permanent deformation)

After subjected to load on rolling bearing, there will be plastic (permanent) deformation on the contact point of the rolling element contacting with the raceway. If the load is excessively heavier, there will be forming a plastic deformation on the raceway surface, which makes rolling bearing producing heavier vibration and noise when rolling bearing rotates.

Foreign hard particles invading into rolling bearing also can form pressing marks. Besides, excessive large impact load, or when rolling bearing is in static condition, machine vibration etc. also could form pit on the contact point. The rolling bearing static load carrying capacity is determined by limiting the permissible permanent deformation value.

(3) Wear

Because of the relative movement between the rolling element and raceway and the invading of dirt and dust, there will be producing wear on the surfaces of the rolling element and raceway. If the wear is excessively large, the bearing clearance, noise and vibration would increase, which therefore lower the rolling bearing running precision, and make direct

influence on the main machine precision. Therefore, for some precision machine bearing, we can use the permissible wear value to limit the bearing life.

The above three rolling bearing failure modes are the most common failures. Besides, there are many other kinds of rolling bearing failures, such as ring fracture, rolling element crush to pieces, cage wear and broke, electric rust by passing current, rust and so on. But the appearance of these failures has accidental casualty related to many factors. Under normal mounting, lubrication, maintainance condition, these failures could be avoided.

The following measures can be used for preventing the three main rolling bearing failures: for fatigue failure, we can use the dynamic rating load to limit bearing load carrying capacity; for plastic deformation, we can use the static rating load to limit bearing static load carrying capacity; for wear, we can use the permissible wear value to limit bearing load carrying capacity.

2.6.2 Basic Regular Pattern of Rolling Bearing Fatigue Life

1. Basic Pattern of rolling bearing fatigue life

A large number of rolling bearing life test data have proved that the rolling bearing fatigue life is considerable dispersion. If a population of apparently identical rolling bearings (identical structure dimensions, material, heat treatment, machining method) is subjected to identical load, speed, lubrication, and environmental conditions, the individual bearings do not achieve the same fatigue life. The lowest life and highest life can be differed from several ten times to even near one hundred times.

Figure 2 – 37 is the endurance test results of a group of 20 rolling bearings. The ordinate expresses the actual life of each bearing, and the abscissa expresses the ordinal number arranged by the actual life value. It can be seen very apparently that the difference between the lowest life and the highest life is very large.

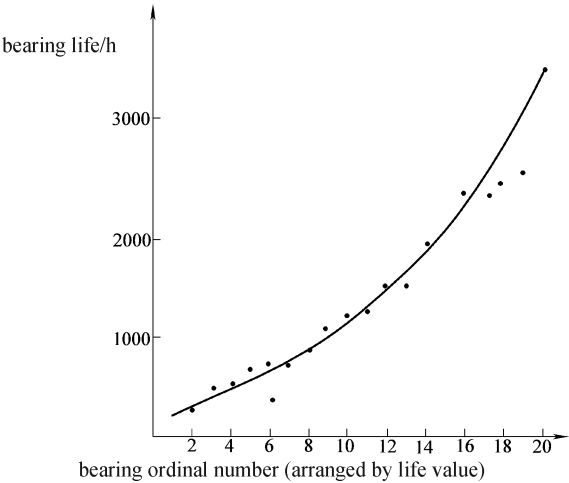


Figure 2 - 37 rolling bearing endurance test result

It can be seen from a large number of rolling bearing endurance test data that most of rolling bearing endurance test results have the same tendency as Figure 2 - 37. Therefore, for rolling bearing fatigue life dispersion situation, Figure 2 - 38 can be the general pattern of rolling bearing fatigue life.

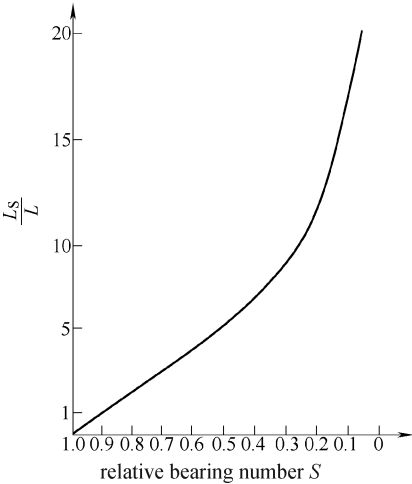


Figure 2 - 38 General pattern of rolling bearing fatigue life

In Figure 2 – 38, the abscissa expresses the relative bearing number S that can achieve at or over the bearing life L_s , and the ordinate expresses the corresponding life L_s to S . (Let the rating life L which 90% bearings can achieve at or over be 1).

From Figure 2 – 38 we can see:

1) For a batch of rolling bearings, the fatigue life that 100% bearings can achieve is zero (for a limited number of bearings, there exists the lowest bearing life).

2) The relative number of bearings having infinite fatigue life is zero. That is any bearing can not rotate for infinite period.

For rolling bearing fatigue life dispersion feature, in order to express a population of rolling bearing fatigue life, we have to use statistical method to calculate the bearing life under certain probability of survival S . It is usual to choose the life of two special points in Figure 2 – 38 to define a group of bearing life, the rating life and the median fatigue life.

Terms related to rolling bearing fatigue life are defined as follows:

1) A rolling bearing fatigue life: this term indicates the total revolutions that this bearing will endure before appearing first fatigue spalling in any rolling element or in any raceway in this bearing, or indicates the hours of successful operation at a given speed.

2) Rating fatigue life L : this term indicates the total revolutions that 90% of an identical population of bearings will endure before appearing fatigue spalling, or indicates the hours of successful operation at a given speed.

3) Median fatigue life L_m : this term indicates the total revolutions that 50% of an identical population of bearings will endure or over before appearing fatigue spalling, or indicates the hours of successful operation at a given speed. The median fatigue life L_m is not the mathematical average value of an identical population of bearing fatigue lives.

Usually, the median fatigue life L_m is about five times of the rating life.

4) Probability of survival S : this term indicates the relative bearing quantity of an identical population of bearings that will endure or over fatigue life L_s . The probability of survival S is described as follows:

$$S = \frac{N_s}{N} \quad (2-163)$$

Where N_s — the number of bearings that have successfully endured L_s revolutions of operation;

N — the total number of bearings.

For a given bearing, the probability of survival S could be the probability that the given bearing will endure or over the fatigue life L_s . Or it is called as reliability.

5) Probability of failure F : this term indicates the relative bearing quantity of an identical population of bearings that will not endure fatigue life L_s . The probability of failure F is described as follows:

$$F = \frac{N - N_s}{N} = 1 - S \quad (2-164)$$

When selecting bearing, the main task is to calculate the rating life under a given working condition. In general, taking $S = 0.9$ is as the rating life. For a bearing, the reliability of surviving the rating life is 90%.

2. Weibull Distribution For Rolling Bearing Fatigue Life

In Figure 2-38, the curve can be seen as an exponential curve, expressed by the following equation:

$$\lg \frac{1}{S} = AL_s^e \quad (2-165)$$

If taking logarithm for both sides of the equation, we can get:

$$\lg \lg \frac{1}{S} = e \lg L_s + \lg A \quad (2-166)$$

Where S — the probability of survival, that is the relative bearing numbers endure or exceed L_s ;

L_s — the bearing life corresponding to S . Usually, using $1-S$ as the corner note. For example, L_{10} expresses the bearing life with 90% probability of survival, and L_5 expresses the life with 95% probability of survival;

e — Weibull slope;

A — constant.

Equation (2-166) is called as rolling bearing fatigue life Weibull distribution. If taking the ordinate as $\lg 1/S$, and the abscissa as L_s on double logarithm paper, let:

$$y = \lg \lg \frac{1}{S}$$

$$x = \lg L_s$$

$$b = \lg A$$

Then the equation will become as:

$$y = ex + b$$

Obviously, the above equation is a straight line equation, The e is the straight line slope. Therefore, the Weibull distribution equation (2-166) is a straight line on double logarithm paper. The e can be calculated from the following equation:

$$e = \lg \frac{\lg \frac{1}{S_2}}{\lg \frac{1}{S_1}} \bigg/ \lg \frac{L_{S2}}{L_{S1}} \quad (2-167)$$

The e value expresses the dispersion degree for an identical population of bearings. The lower value of e indicates greater dispersion of fatigue life of the identical population bearings. Therefore the e value also can be called as dispersion exponent. Usually, the e values are as follows:

For ball bearing $e = 10/9$;

For roller bearing $e = 9/8$.

The two straight lines in Figure 2-39 are based on the actual bearing endurance data from bearings fabricated by Sweden SKF bearing company.

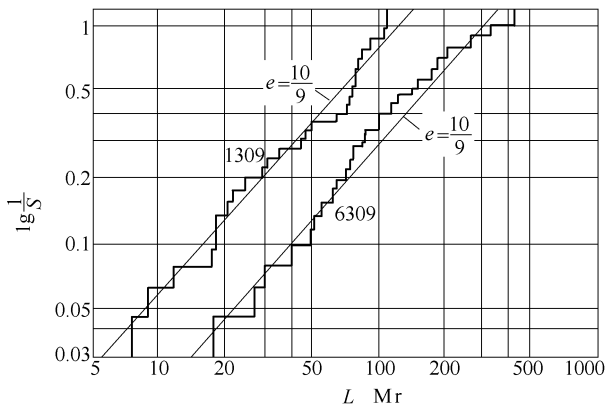


Figure 2-39 Rolling bearing fatigue life Weibull distribution

Using the equation (2-166), we can calculate the bearing fatigue life under any probability of survival.

In the equation (2-166), if taking $S=0.9$, then $L_s=L$, so we can get:

$$A = \frac{\lg \frac{1}{0.9}}{L^e}$$

Putting this into the equation (2-166), we can get:

$$\lg \frac{1}{S} = \left(\frac{L_s}{L} \right)^e \lg \frac{1}{0.9} = 0.0457 \left(\frac{L_s}{L} \right)^e \quad (2-168)$$

If knowing the rating life L and Weibull slope e in the equation (2-168), then we can easily get the life under any probability of survival.

After analyzing the 2250 bearing endurance test data, it is confirmed by T. Tallian that bearing fatigue life conforms with Weibull distribution within $0.4 < S < 0.93$ scope. So the equation (2-168) can use within $0.4 < S < 0.93$ scope. The rolling bearing rating load theory is based on that bearing fatigue life conforms with Weibull distribution.

There are two applications for the equation (2-168):

1) Usually, while selecting bearing, first of all, we calculate the bearing life with reliability 90% (that is the rating life L) according to the rating load data in the bearing manufacturer catalog. If according to the actual need that the bearing reliability is not 90%, then according to the equation (2-168), we can calculate the bearing life of any reliability. (The bearing life calculation method of reliability that is higher than 93% is in Section 2.7 of this chapter).

[**Example 2-10**] According to bearing working condition, the 6307 bearing rating life should be 6000h. If the required reliability is as $S=0.8$, how much the bearing life is?

[**Resolve**] Putting $S=0.8$, $L=6000$, $e=10/9$ into the equation (2-168), we can get:

$$\lg \frac{1}{0.8} = 0.0457 \left(\frac{L_{20}}{6000} \right)^{10/9}$$

$$L_{20} = \left(\frac{\lg \frac{1}{0.8}}{0.0457} \right)^{9/10} * 6000 = 11750 \text{h}$$

2) When there are many sets of the same type machines running at the same time, after having been running certain hours, the running time of the remaining bearings can be estimated according to the numbers of damaged bearings.

[**Example 2 – 11**] There are 100 spherical roller bearings running under given working condition. After having been running 2000h, 20 bearings have been damaged. To estimate how many bearings will be damaged after continuing running another 1000h?

[**Resolve**] After 20 bearings having been damaged, there are still 80 bearings continuing running, so the probability of survival after running 2000h is:

$$S = (100 - 20)/100 = 80\%$$

Assume that the the probability of survival after running another 1000h is S_1 ,

$$\text{Then } L_{s1} = (2000 + 1000) \text{ h} = 3000 \text{h}$$

For roller bearing taking: $e = 9/8$.

From the equation (2 – 168) we can get:

$$\lg \frac{1}{0.8} = 0.0457 * \left(\frac{2000}{L} \right)^{9/8}$$

$$\lg \frac{1}{S_1} = 0.0457 * \left(\frac{3000}{L} \right)^{9/8}$$

From the above equations we can get:

$$\lg \frac{1}{S_1} = \left(\frac{3000}{2000} \right)^{9/8} * \lg \frac{1}{0.8} = 0.153$$

$$S_1 = 0.70$$

Therefore, after continuing running another 1000h, there will be another 10 bearings damaged.

[**Example 2 – 12**] There are a group of ball bearings running under the same working condition in a work shop. Their rating life is as $L = 1000 \text{h}$. After having been running 1500h, some bearings have been dam-

aged. If the 90% bearings of the remaining bearings can continue running undamaged, estimate for how long these bearings could also be continuing running?

[**Resolve**] Assume: S_a is the relative bearing number of undamaged bearings after having been running $L_a = 1500\text{h}$, then the relative bearing number of 90% undamaged bearings of the remaining bearings is $S_b = 0.9 S_a$, the relative life with S_b is as L_b .

From the equation (2-168) we can get:

$$\lg \frac{1}{S_a} = 0.0457 \left(\frac{L_a}{L} \right)^{10/9}$$

$$\lg \frac{1}{S_b} = \lg \frac{1}{0.9 S_a} = 0.0457 \left(\frac{L_b}{L} \right)^{10/9}$$

But

$$\lg \frac{1}{0.9 S_a} = \lg \frac{1}{0.9} + \lg \frac{1}{S_a}$$

Therefore, from the above two equations we can get:

$$\begin{aligned} L_b &= (L_a^{10/9} + L^{10/9})^{9/10} = (1500^{10/9} + 1000^{10/9})^{9/10} \text{h} \\ &= 2340\text{h} \\ 2340\text{h} - 1500\text{h} &= 840\text{h} \end{aligned}$$

So, after having been running 1500h, 90% bearings of the remaining bearings can continue running 840h undamaged.

2.6.3 Basic Dynamic Load Rating of Rolling Bearing

1. Basic principle to determine rolling bearing basic dynamic load rating

(1) Basic hypotheses for statistical method of rolling bearing contact fatigue problem

Knowing from the above section that rolling bearing fatigue lives are very dispersion. “The reason of a matter development is in matter interior, but not in matter exterior, and it is in the matter interior contradictory characteristic”. For everything of a group of rolling bearings looks the same in the macroscopic view, their fatigue lives are very dispersion. The reason must be found from material interior.

On the basis of a large amount of test data, for dealing with material strength by statistical method, Weibull established a statistical material

strength theory. But the Weibull theory studied a static failure of brittle engineering materials. The theory is based on the assumption that an initial crack leads to a break immediately. For the fatigue life of rolling bearings, there is no crack on the contact surface from the macroscopic view. But experience has demonstrated that many cracks are formed below the surface that do not propagate to the surface. Thus, Weibull's theory is not directly applicable to solving rolling bearing fatigue life problem.

On the basis of Weibull's theory, G. Lundberg and A. Palmgren have established their own rolling bearing rating load theory. The basic hypotheses are as follows:

1) Rolling bearing material is not homogeneous. There exist some strength weak points in it. At the places of having non-metal slag and metallurgical crystal structure defect, there have been forming material strength weak points. While load is applied, there will be stress concentration around the material strength weak points. After repeated loading, there will be plastic deformation and sliding around the material strength weak points. At last there will be forming interior cracks in neighboring area of the material strength weak points. Under continuing running, the interior cracks will be propagated to the contact surface, forming metal surface spalling.

The decisive role for rolling bearing contact fatigue is the contact body interior shear stress τ_{zy} . It is parallel to the rolling direction and the maximum shear stress occurs at the depth Z_0 below the surface. During the passage of a loaded rolling element rolling over a point on the raceway surface, the shear stress τ_{zy} on the z axis varies between $\pm \tau_0$. Therefore, while dealing with the contact fatigue problem, the probability of failure of material fatigue is assumed to be related to the τ_0 . The greater the τ_0 is, the greater the probability of failure of material fatigue is, and the (smaller the) probability of survival of material fatigue is.

2) Owing to the time required for a crack to propagate from the subsurface depth of initiation to the surface, while the initiation crack occurs, the material failure does not occur at once. There is a process

during which the initial crack propagates to the surface and forms a spalling. Thus, the probability of failure of material fatigue is related to the depth Z_0 of the maximum shear stress $\tau_{z_{\max}} = \tau_0$. The greater the depth Z_0 is, the smaller the probability of failure of material fatigue is.

3) The failure of material fatigue is produced gradually. Thus, the probability of failure of material fatigue is proportional to the stress cycling number N . The greater the N is, the greater the probability of the failure of material fatigue is.

4) For a contact body subjecting to load, the most dangerous failure area is only in the neighboring area of the maximum stress area. Thus, the probability of failure of material fatigue is related to the numbers of the material strength weak point. Within the stressed volume, the greater the material strength weak point number is, the probability of failure of material fatigue is. From statistical view, a large volume includes more is weak points. Thus the size of the stressed volume is evidently a measure of the numbers of weak points under stress. That is: the greater the stressed volume is, the greater the probability of failure of material fatigue is.

on the basis of the above mentioned hypotheses, G. Lundberg and A. Palmgren postulated an exponential equation for statistical approach dealing with the contact fatigue problem as follows:

$$\lg \frac{1}{S} \propto \frac{\tau_0^c N^e}{z_0^h} V \quad (2-169)$$

Where S — Probability of survival of material fatigue enduring N (million) stress cycles;

τ_0 — Magnitude of the subsurface shear stress paralleling with rolling direction;

Z_0 — Depth of the shear stress τ_0 ;

V — Volume under stress;

e, c, h — Exponents to be determined; e is Weibull distribution slope.

In a rolling bearing, rolling element contacts with inner ring, and also contacts with outer ring. So, it is more complicated. For simplification, it can be approximately assumed that the probability of failure of rolling

element contacting with inner ring is as same as the probability of failure of the inner ring, and the probability of failure of rolling element contacting with outer ring is as same as the probability of failure of the outer ring. In the following discussion, the probability of failure of inner ring and outer ring will be discussed.

(2) Relationship between rolling element rating load and bearing life
Magnitude of shear stress τ_0 and Z_0 :

Experiment has demonstrated that during rolling, usually the fatigue crack occurs in the plane paralleling with the rolling direction. Knowing from Section 2.2.4:

$$\tau_0 = \frac{1}{2} T \sigma_{\max}$$

$$z_0 = \zeta b \quad (2-39)$$

$$T = \frac{\sqrt{2t-1}}{t(t+1)} \quad (2-37)$$

$$\zeta = \frac{1}{(t+1)\sqrt{2t-1}} \quad (2-40)$$

$$\frac{b}{a} = \sqrt{(t^2-1)(2t-1)} \quad (2-38)$$

Where $\sigma_{\max}$ — Maximum compressive stress on contact surface;

T, ζ — Function of b/a ;

t — Auxiliary parameter;

Values of $t, b/a, T$ can be checked out from Table 2-36.

From Hertz theory we can get:

$$\sigma_{\max} = \frac{3Q}{2\pi ab} \quad (2-22)$$

$$a = \mu \sqrt[3]{\frac{3Q}{E_0 \Sigma \rho}} \quad (2-170)$$

$$b = \nu \sqrt[3]{\frac{3Q}{E_0 \Sigma \rho}} \quad (2-171)$$

$$E_0 = \frac{E}{1 - \frac{1}{m^2}} \quad (2-172)$$

Where Q — Rolling bearing load (N);

$\sum \rho$ — Curvature sum;

E — Material modulus of elasticity;

$1/m$ — Poisson's ratio;

μ, ν — Factors determining the semi-major axis and semi-minor axis of contact ellipse. Related to function $F(\rho)$.

Here, it is assumed that the materials of two contact bodies are the same, taking the same value of the E and $1/m$.

1) Volume V under stress

For volume under stress, G. Lundberg and A. Palmgren postulated that

$$V \propto a z_0 l \quad (2-173)$$

where l — Raceway length.

Obviously, the three measures of a , Z_0 and l designate the stress expanding in width, depth and length of contact body.

For point contact, we can take:

$$V \propto \pi D_n a z_0 \quad (2-174)$$

For line contact, we can take:

$$V \propto \frac{3}{4} \pi D_n l_{we} z_0 \quad (2-175)$$

Where l_{we} — Effective contact length of a roller;

D_n — Raceway diameter on contact place.

2) Stress cycling number N

N is the stress cycling number, and its unit is in million numbers. L designates bearing life (its unit is million numbers), and u designates rolling contact number. Then

$$N \propto uL$$

Putting τ_0 , z_0 , V and N into the equation (2-169), and introducing rolling element diameter, letting:

$$D_w^{\frac{c+h-1}{2}} D_w^{\frac{c-h-1}{2}} \left(\frac{1}{D_w^2} \right)^{\frac{c-h+1}{2}} D_w^{2-h} = 1$$

We can get:

$$\lg \frac{1}{S} \propto \frac{T^e}{\zeta^{h-1}} \left[\frac{E_0 D_w \Sigma \rho}{3 \mu v^2} \right]^{\frac{c+h-1}{2}} \left[\frac{D_w}{a} \right]^{\frac{c-h-1}{2}} \\ * \left[\frac{Q}{D_w^2} \right]^{\frac{c-h+1}{2}} D_n D_w^{2-h} u^e L^e \quad (2-176)$$

Equation (2-176) is suitable for both point contact as well as line contact.

For point contact, putting Equation (2-170) into the equation (2-176) can get:

$$\lg \frac{1}{S} \propto \frac{T^e}{\zeta^{h-1} \mu^{c-1} v^{c+h-1}} \left[\frac{E_0 D_w \Sigma \rho}{3} \right]^{\frac{2c+h-2}{3}} \\ * \left[\frac{Q}{D_w^2} \right]^{\frac{c-h+2}{3}} D_n D_w^{2-h} u^e L^e \quad (2-177)$$

For line contact, because of the b/a value is very small, if taking a limit, we can get:

$$\lim_{\frac{b}{a} \rightarrow 0} \mu v^2 = \frac{2}{\pi}$$

When $b/a = 0$, designating the T , ζ as T_0 , ζ_0 , we can get:

$$\lg \frac{1}{S} \sim \frac{T_0^e}{\zeta_0^{h-1}} \left[\frac{\pi E_0 D_w \Sigma \rho}{6} \right]^{\frac{c+h-1}{2}} \left[\frac{4}{3} \frac{D_w}{l_{we}} \right]^{\frac{c-h-1}{2}} \\ * \left[\frac{Q}{D_w^2} \right]^{\frac{c-h+1}{2}} D_n D_w^{2-h} u^e L^e \quad (2-178)$$

In actual application, usually S is a constant, in determining bearing rating load, taking $S = 0.9$.

For point contact, we can rewrite the equation (2-177) as:

$$QL^{\frac{3e}{c-h+2}} = A_1 \phi D_w^{\frac{2c+h-5}{c-h+2}} \quad (2-179)$$

$$\phi = \left\{ \left[\frac{T}{T_1} \right]^e \left[\frac{\zeta_1}{\zeta} \right]^{h-1} \frac{D_w \Sigma \rho}{\mu^{c-1} v^{c+h-1}} * \frac{D_n}{D_w} u^e \right\}^{-\frac{3}{c-h+2}} \quad (2-180)$$

Where T_1 , ζ_1 are the values of T , ζ while $b/a = 1$; A_1 is a constant.

The rolling element load while $L = 1$ is called as the rating rolling element load, expressed as Q_c . We can get from the equation (2-179):

$$Q_c = A_1 \Phi D_w^{\frac{2c+h-5}{c-h+2}} \quad (2-181)$$

From the equations (2-179) and (2-181), we can get the relationship between rolling element load and bearing life for point contact:

$$L = \left(\frac{Q_c}{Q} \right)^{\frac{c-h+2}{3e}} \quad (2-182)$$

For line contact, we can rewrite the equation (2-178) as:

$$QL^{\frac{2e}{c-h+1}} = B_1 \psi D_w^{\frac{c+h-3}{c-h+1}} l_{we}^{\frac{c-h-1}{c-h+1}} \quad (2-183)$$

When taking $L = 1$, we can get the rating rolling element load as:

$$Q_c = B_1 \psi D_w^{\frac{c+h-3}{c-h+1}} l_{we}^{\frac{c-h-1}{c-h+1}} \quad (2-184)$$

$$\begin{aligned} \text{Where } B_1 = & \left(\frac{3}{4} \right)^{\frac{3-h-1}{c-h+1}} \left(\frac{\pi}{2} \right)^{-\frac{c+h-1}{c-h+1}} \left[\frac{T_1}{T_0} \right]^{\frac{2e}{c-h+1}} \\ & * \left[\frac{\zeta_0}{\zeta_1} \right]^{\frac{2(h-1)}{c-h+1}} \left(\frac{E_0}{3} \right)^{\frac{c-h-1}{3(c-h+1)}} A_1^{\frac{2(c-h+2)}{3(c-h+1)}} \end{aligned} \quad (2-185)$$

$$\psi = \left\{ [D_w \Sigma \rho]^{\frac{c+h-1}{2}} \frac{D_n}{D_w} u^e \right\}^{\frac{2}{c-h+1}} \quad (2-186)$$

From the equations (2-183) and (2-184), we can get:

$$L = \left(\frac{Q_c}{Q} \right)^{\frac{c-h+1}{2e}} \quad (2-187)$$

Rewriting the equations (2-182) and (2-187) as a unifying form, we can get:

$$L = \left(\frac{Q_c}{Q} \right)^e \quad (2-188)$$

For point contact:

$$\varepsilon = \frac{c-h+2}{3e} \quad (2-189)$$

For line contact:

$$\varepsilon = \frac{c-h+1}{2e} \quad (2-190)$$

(3) Determine the exponents in bearing life equation

The exponents in the rolling bearing life calculation equation (2-188) were determined by a large amount of test data.

1) Weibull distribution slope e

The e expresses the material strength dispersion characteristic. Therefore, for different type of rolling bearings made by the same material,

the e value is nearly the same. Through a large amount of test data, we can get statistical value of e that is nearly 1.1 or so. In order to deduce conveniently, usually taking $e = 10/9$ for point contact, and taking $e = 9/8$ for line contact.

2) Rolling element diameter D_w exponent

In order to determine the D_w exponent, the 1500 sets of ball bearing of diameter $D_w = 1.5 \sim 25\text{mm}$ range have been tested. The D_w exponent is obtained as 1.8, that is in the equation (2-179), the D_w exponent is:

$$\frac{2c+h-5}{c-h+2} = 1.8$$

3) Bearing life exponent ε

Through a large amount of test data, it is proved that for ball bearing we can take bearing life exponent as:

$$\varepsilon = 3$$

Through 180 sets of double row spherical roller bearing test data, it is proved that for roller bearing we can take bearing life exponent as:

$$\varepsilon = 4$$

4) From having been knowing exponents of e , ε and D_w , we can calculate out other exponents as listed in Table 2-16.

Table 2-16 Exponent values

contact type	e	c	h	ε	D_w exponent	L_y exponent
point contact	$\frac{10}{9}$	$\frac{31}{3}$	$\frac{7}{3}$	3	1.8	—
line contact	$\frac{9}{8}$	$\frac{31}{3}$	$\frac{7}{3}$	4	$\frac{29}{27}$	$\frac{7}{9}$

Putting the values of Table 2-16 into the equations (2-180), (2-181) and (2-184) to (2-186), we can get:

$$\text{For point contact: } Q_c = A_1 \phi D_w^{1.8} \quad (2-191)$$

$$\phi = \left[\frac{T_1}{T} \right]^{3.1} \left[\frac{\zeta}{\zeta_1} \right]^{0.4} \frac{\mu^{2.8} \nu^{3.5}}{(D_w \Sigma \rho)^{2.1}} \left[\frac{D_w}{D_n} \right]^{0.3} u^{-\frac{1}{3}} \quad (2-192)$$

For line contact:

$$Q_c = B_1 \psi D_w^{\frac{29}{27}} l_{we}^{\frac{7}{9}} \quad (2-193)$$

$$\psi = (D_w \Sigma \rho)^{-\frac{35}{27}} \left[\frac{D_w}{D_n} \right]^{\frac{2}{9}} u^{-\frac{1}{4}} \quad (2-194)$$

$$B_1 = 3.517 A_1^{\frac{20}{27}} \quad (2-195)$$

(4) Rating rolling element load Q_c

Introducing the function (ρ) related to contact surface shape

$$F(\rho) = \frac{(\rho_{11} - \rho_{12}) + (\rho_{11} - \rho_{12})}{\Sigma \rho} \quad (2-19)$$

Figure 2-40 and Figure 2-41 designate the methods for ball bearing and spherical roller curvature radius.

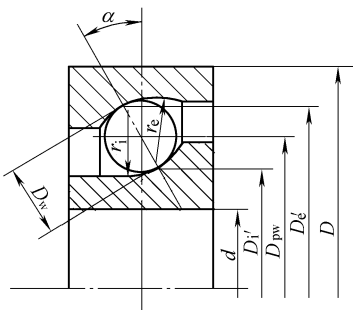


Figure 2-40 Ball bearing

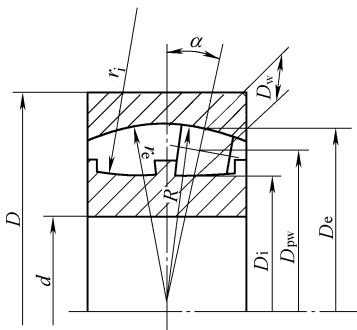


Figure 2-41 Spherical roller bearing

Where $\rho_{11} = \frac{2}{D_w}$ $\rho_{12} = \frac{1}{R}$ (spherical roller bearing)

$\rho_{12} = \frac{2}{D_w}$ (ball bearing)

$\rho_{11} = \pm \frac{2 \cos \alpha}{D_n}$ $\rho_{12} = -\frac{1}{r}$

The relationship between raceway diameter and rolling element center circle pitch diameter is:

$$D_n = D_i = D_{pw} - D_w \cos \alpha \quad (2-15)$$

$$D_n = D_e = D_{pw} + D_w \cos \alpha \quad (2-16)$$

$$\gamma = \frac{D_w \cos \alpha}{D_{pw}} \quad (2-17)$$

Therefore we can get:

$$[1 + F(\rho)] \frac{D_w}{2} \Sigma \rho = \frac{2}{1 \mp \gamma} \quad (2-196)$$

$$[1 - F(\rho)] \frac{D_w}{2} \Sigma \rho = D_w \left[\frac{1}{R} - \frac{1}{r} \right] \quad (2-197)$$

In a bearing, inner and outer ring contact numbers are:

$$u = \frac{z}{2} [1 \pm \gamma] \quad (2-147), (2-148)$$

Putting the equations (2-196) and (2-197), (2-147), (2-148) into the equations (2-192) and (2-194), we can get:

$$\phi = 0.06855 \omega \frac{[1 \mp \gamma]^{1.8}}{[1 \pm \gamma]^{\frac{1}{3}}} \left[\frac{D_w}{D_{pw}} \right]^{0.3} z^{-\frac{1}{3}} \quad (2-198)$$

$$\text{In which } \omega = [1 + F(\rho)]^{2.1} \left[\frac{T_1}{T} \right]^{3.1} \left[\frac{\xi}{\xi_1} \right]^{0.4} \mu^{2.8} \nu^{3.5} \quad (2-199)$$

For line contact, $F(\rho) = 1$, thus we can get:

$$\psi = 0.4842 \frac{[1 \mp \gamma]^{\frac{29}{27}}}{[1 \pm \gamma]^{\frac{1}{4}}} \left[\frac{D_w}{D_{pw}} \right]^{\frac{2}{9}} z^{-\frac{1}{4}} \quad (2-200)$$

Introducing the function:

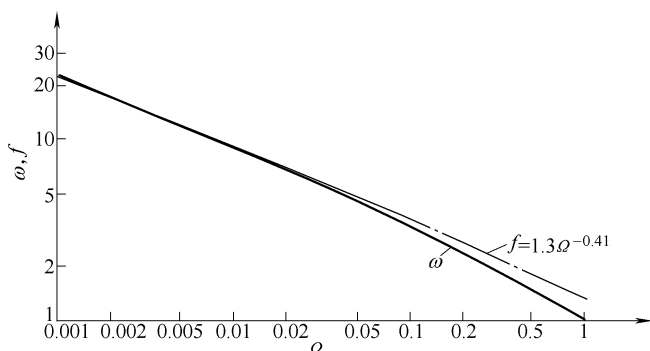
$$\Omega = \frac{1 - F(\rho)}{1 + F(\rho)} = \frac{D_w}{2R} \cdot \frac{(r - R)}{r} [1 \mp \gamma]$$

Basis on Lundberg and Palmgren assumption, introducing:

$$f = 1.3 \Omega^{-0.41}$$

As shown in Figure 2-42, then:

$$\begin{aligned} \omega &\approx f = 1.3 \Omega^{-0.41} \\ &= 1.3 \left[\frac{2R}{D_w} \cdot \frac{r}{(r - R)} \right]^{0.41} [1 \mp \gamma]^{-0.41} \end{aligned} \quad (2-201)$$

Figure 2-42 Function ω and Ω relation

Putting the equations (2-198), (2-200) and (2-201) into the equations (2-191) and (2-193), we can get the rating rolling element load calculation equation as:

For point contact:

$$Q_c = A \left[\frac{2R}{D_w} \frac{r}{r-R} \right]^{0.41} \frac{[1 \mp \gamma]^{1.39}}{[1 \pm \gamma]^{\frac{1}{4}}} \left[\frac{D_w}{D_{pw}} \right]^{0.3} * D_w^{1.8} z^{-\frac{1}{3}} \quad (2-202)$$

For line contact:

$$Q_c = B \frac{[1 \mp \gamma]^{\frac{29}{27}}}{[1 \pm \gamma]^{\frac{1}{4}}} \left[\frac{D_w}{D_{pw}} \right]^{\frac{2}{9}} D_w^{\frac{29}{27}} l_w^{\frac{7}{9}} z^{-\frac{1}{4}} \quad (2-203)$$

For the above equations, the upper sign is suitable for inner ring, and the lower sign is suitable for outer ring.

For bearings made by bearing steel and with the hardness conforming to the standard, a large amount of test can get:

$$A = 98 \quad B = 551$$

(5) Average rolling element load Q_i , Q_e

Only under pure central axial load, every rolling element in a bearing is subjecting to the same load. Under radial load or combined load of radial load and axial load, every rolling element in a bearing is not subjecting to the same value. Therefore, when determining rating rolling element load, a concept of “average” load has to be introduced.

For easily derivation, we assume that the inner ring is rotating, and the

outer ring is in stationary.

According to analysis for load distribution in a bearing, the rolling element load is related to rolling element position in a bearing:

$$Q_{\psi} = Q_{\max} \left[1 - \frac{1}{2T} (1 - \cos\psi) \right]^t \quad (2-77)$$

For point contact: $t = 3/2$

For line contact: $t = 1.1$

While inner ring is rotating, every inner ring point is passing through loading area, subjecting to the same varying load. According to the theory of fatigue accumulated damage, the average rolling element load has the cubic power of the average load for point contact, and the fourth power of the average load for line contact.

If Q_i designates the inner ring average rolling element load, then

$$Q_i = \left[\frac{1}{2\pi} \int Q^s(\psi) d\psi \right]^{\frac{1}{s}} \quad (2-204)$$

For point contact $s = 3$

For line contact $s = 4$

Putting the equation (2-77) into the equation (2-204), we can get:

$$Q_i = Q_{\max} J_1 \quad (2-205)$$

$$J_1 = \left\{ \frac{1}{2\pi} \int_{-\psi_0}^{+\psi_0} \left[1 - \frac{1}{2T} (1 - \cos\psi) \right]^s d\psi \right\}^{\frac{1}{s}} \quad (2-206)$$

For point contact $st = 3 * \frac{3}{2} = \frac{9}{2}$

For line contact $st = 4 * 1.1 = 4.4$

While inner ring is rotating, and outer ring is in stationary, every point of the outer ring is subjecting to non-changing load. But its value is related to its position on the circle. The probability of survival of the entire outer ring equals to the product of the probabilities of survivals of every part of the outer ring.

For a given bearing, there are only variables of Q and L in the equations (2-177) and (2-178), therefore, we can rewrite the equations (2-177) and (2-178) as:

$$\lg \frac{1}{S} \propto Q^w L^e \quad (2-207)$$

For point contact: $w = \frac{C-h+2}{3} = \frac{10}{3}$

For line contact:
$$w = \frac{C - h + 1}{2} = \frac{9}{2}$$

Equation (2-207) fits every part of the outer ring, thus for the entire outer ring:

$$lg \frac{1}{S_e} \propto \int Q^w(\psi) L^e \frac{D_e}{2} d\psi = Q_e^w L^e \pi D_e \quad (2-208)$$

Where S_e — entire outer ring probability of survival;

Q_e — outer ring average rolling element load.

$$Q_e = \left[\frac{1}{2\pi} \int_{-\psi_0}^{+\psi_0} Q^w(\psi) d\psi \right]^{\frac{1}{w}} = Q_{\max} J_2 \quad (2-209)$$

$$J_2 = \left\{ \frac{1}{2\pi} \int_{-\psi_0}^{+\psi_0} \left[1 - \frac{1}{2T} (1 - \cos t\psi) \right]^{wt} d\psi \right\}^{\frac{1}{w}} \quad (2-210)$$

For point contact
$$wt = \frac{10}{3} * \frac{3}{2} = 5$$

For line contact
$$wt = 9/2 * 1.1 = 4.95$$

From the equations (2-206) and (2-210), J_1 and J_2 are the function related to load distribution parameter T, listed in Table 2-17.

Table 2-17 Single row bearing J_1, J_2

T	point contact		line contact	
	J_1	J_2	J_1	J_2
0	0	0	0	0
0.1	0.4275	0.4608	0.5287	0.5633
0.2	0.4806	0.5100	0.5772	0.6073
0.3	0.5150	0.5427	0.6079	0.6359
0.4	0.5411	0.5673	0.6309	0.6571
0.5	0.5625	0.5875	0.6495	0.6744
0.6	0.5808	0.6045	0.6653	0.6888
0.7	0.5970	0.6196	0.6792	0.7015
0.8	0.6104	0.6330	0.6906	0.7127
0.9	0.6248	0.6453	0.7028	0.7229
1.0	0.6372	0.6566	0.7132	0.7323
1.25	0.6652	0.6821	0.7366	0.7532
1.67	0.7064	0.7190	0.7705	0.7832
2.5	0.7707	0.7777	0.8216	0.8301
5	0.8675	0.8693	0.8989	0.9014
∞	1	1	1	1

The values in Table 2 – 17 are calculated out by J_1 (9/2; 4) instead of J_1 (4.4; 4), and by J_2 (5; 9/2) instead of J_2 (4.95; 9/2). The error is very little, and can be omitted.

The corner note 1 expresses the rotating ring, and the corner note 2 expresses the stationary ring.

From the load distribution theory, we can get:

$$Q_{\max} = \frac{F_r}{z \cos \alpha J_r} \quad (2-84)$$

Putting the equation (2-84) into the equations (2-205) and (2-209) we can get:

$$Q_i = Q_{\max} J_1 = \frac{F_r}{z \cos \alpha} \frac{J_1}{J_r} \quad (2-211)$$

$$Q_e = Q_{\max} J_2 = \frac{F_r}{z \cos \alpha} \frac{J_2}{J_r} \quad (2-212)$$

2. Determination method of rolling bearing basic dynamic load rating

(1) Definition

The rolling bearing dynamic load rating is the bearing load carrying capacity determined under the assumed running conditions. These assumed running conditions are as follows:

1) Inner ring rotates relative to load vector, and outer ring is in stationary relative to the load vector.

2) Probability of survival $S = 90\%$.

3) Basic rating life $L = 1$ (million revolutions).

4) For radial and angular contact bearing, it is assumed that there is only pure radial displacement for inner ring and outer ring, that is there is only half-raceway subjecting to load, and the load distribution parameter $T = 0.5$. For thrust and thrust angular contact bearing, it is assumed that there is only pure axial displacement, that is there is only central axial load.

From the above assumptions, the basic dynamic load rating can be defined as a radial load for which 90% of a group of apparently identical bearings will survive or over for one million revolutions with inner ring rotating and outer ring in stationary. For radial and angular radial bearing, the basic dynamic load rating is a component of the radial load under which

only half raceway is subjecting to load. For thrust and thrust angular bearing, the basic dynamic load rating is a central axial load.

(2) Inner ring load rating C_i , outer ring load rating C_e

In a rolling bearing, rolling element contacts with both inner ring and outer ring. Therefore first of all, the inner ring load rating C_i and the outer ring load rating C_e are determined from the rolling element load rating Q_c . Then the bearing basic dynamic load rating C is calculated from the product law of probability.

In the equations (2-211) and (2-212), if Q_i and Q_e denotes the inner ring and outer ring rolling element load rating Q_{ci} and Q_{ce} respectively, then in these equations, F_r corresponds to the inner ring and outer ring load rating C_i and C_e respectively.

That is:

$$C_i = Q_{ci} z \cos \alpha \frac{J_r(0.5)}{J_1(0.5)} \quad (2-213)$$

$$C_e = Q_{ce} z \cos \alpha \frac{J_r(0.5)}{J_2(0.5)} \quad (2-214)$$

From Tables (2-12) and (2-17) we can check out:

For point contact: $J_r(0.5) = 0.2288$ $J_1(0.5) = 0.5625$

$J_2(0.5) = 0.5875$

For line contact: $J_r(0.5) = 0.2453$ $J_{11}(0.5) = 0.6512$

$J_2(0.5) = 0.6750$

(Note: Here $J_1(0.5)$ and $J_2(0.5)$ are obtained from precise calculation)

Putting values of $J_r(0.5)$, $J_1(0.5)$ and $J_2(0.5)$ into the equations (2-213) and (2-214), we can get:

For point contact: $C_i = 0.407 Q_{ci} z \cos \alpha$ (2-215)

$C_e = 0.389 Q_{ce} z \cos \alpha$ (2-216)

For line contact: $C_i = 0.3767 Q_{ci} z \cos \alpha$ (2-217)

$C_e = 0.3634 Q_{ce} z \cos \alpha$ (2-218)

(3) Rolling bearing basic dynamic load rating C

Knowing from the bearing load distribution analysis that rolling element load is in direct proportion to radial load F_r , we can rewrite the equation (2-207) as:

$$\lg \frac{1}{S} \propto F_r^w L^e \quad (2-219)$$

If S_i denotes the inner ring probability of survival, S_e denotes the outer ring probability of survival, and S denotes the entire bearing probability of survival, then for inner ring we can get:

$$\lg \frac{1}{S_i} = k_i F_r^w L^e \quad (2-220)$$

If taking $S_i = 0.9 \quad L = 1$

Then $F_r = C_i$

So $k_i = \lg \frac{1}{0.9} / C_i^w$

For outer ring we can get:

$$\lg \frac{1}{S_e} = k_e F_r^w L^e \quad (2-221)$$

If taking $S_e = 0.9 \quad L = 1$

Then $F_r = C_e$

So $k_e = \lg \frac{1}{0.9} / C_e^w$

For entire bearing:

$$S = S_i S_e$$

$$\lg \frac{1}{S} = \lg \frac{1}{S_i S_e} = (k_i + k_e) F_r^w L^e \quad (2-222)$$

If taking $S = 0.9 \quad L = 1$

Then $F_r = C$

From the equations (2-220), (2-221), (2-222) we can get:

$$\frac{1}{C^w} = \frac{1}{C_i^w} + \frac{1}{C_e^w} \quad (2-223)$$

$$C = g_c C_i \quad (2-224)$$

$$g_c = \left[1 + \left(\frac{C_i}{C_e} \right)^w \right]^{-\frac{1}{w}} \quad (2-225)$$

(4) Multiple-row bearing basic dynamic load rating

If there are i row rolling elements in a bearing, the entire bearing probability of survival is the product of every bearing probability of survival.

Using 1, 2, 3, ... i expresses every row; $C_1, C_2, C_3 \dots$ express every row

bearing basic dynamic load rating; F_r expresses entire bearing subjected radial load. If assuming that the subjected load of every row bearing is the same value, then the load subjected by every row bearing is as F_i , from the equation (2-219) we can get

$$\lg \frac{1}{S_1} = k_1 \left(\frac{F_r}{i} \right)^w L^e$$

$$\lg \frac{1}{S_2} = k_2 \left(\frac{F_r}{i} \right)^w L^e$$

$$\lg \frac{1}{S_3} = k_3 \left(\frac{F_r}{i} \right)^w L^e$$

$$\vdots$$

But

$$\lg \frac{1}{S} = k F_r^w L^e$$

If taking

$$S_1 = S_2 = S_3 = \dots = S = 0.9$$

$$L = 1$$

The constant in every equation can be calculated from each of the following equations:

$$k_1 = \lg \frac{1}{0.9} / C_1^w$$

$$k_2 = \lg \frac{1}{0.9} / C_2^w$$

$$k_3 = \lg \frac{1}{0.9} / C_3^w$$

$$\vdots$$

$$k = \lg \frac{1}{0.9} / C^w$$

From the law of Probability product we can get:

$$S = S_1 S_2 S_3 \dots S_i$$

$$\begin{aligned} \text{Therefore } \lg \frac{1}{S} &= \frac{\lg \frac{1}{0.9}}{C_1^w} \left(\frac{F_r}{i} \right)^w L^e + \frac{\lg \frac{1}{0.9}}{C_2^w} \left(\frac{F_r}{i} \right)^w L^e \\ &+ \frac{\lg \frac{1}{0.9}}{C_3^w} \left(\frac{F_r}{i} \right)^w L^e + \dots = \frac{\lg \frac{1}{0.9}}{C^w} F_r^w L^e \end{aligned}$$

From that we can deduce:

$$\frac{i^w}{C^w} = \frac{1}{C_1^w} + \frac{1}{C_2^w} + \frac{1}{C_3^w} + \dots + \frac{1}{C_i^w}$$

If the bearing basic dynamic load rating of every row bearing is the same, that is:

$$C_1 = C_2 = C_3 = \dots$$

Then we can get:

$$C = i^{1-\frac{1}{w}} C_1 \quad (2-226)$$

For point contact:

$$1 - \frac{1}{w} = 1 - \frac{3}{10} = 0.7$$

$$\text{For line contact:} \quad 1 - \frac{1}{w} = 1 - \frac{2}{9} = \frac{7}{9}$$

So, the double row bearing basic dynamic load rating is not twice of the single row basic dynamic load rating, but for point contact is $2^{0.7} = 1.62$ times, and for line contact is $2^{7/9} = 1.71$ times.

From the equations (2-224) and (2-226) we can get the general expression of bearing basic dynamic load rating as:

$$C = g_c i^{1-\frac{1}{w}} C_i \quad (2-227)$$

3. Ball bearing basic dynamic load rating

The above mentioned point contact equations, all are suitable for ball bearings.

(1) Radial and angular contact ball bearing basic dynamic load rating

Putting the equations (2-202), (2-215), (2-216) into the equation (2-224) we can get:

$$C = 39.9 g_c \left[\frac{2r_i}{2r_i - D_w} \right]^{0.41} \frac{[1 - \gamma]^{1.39}}{[1 + \gamma]^{1/3}} \gamma^{0.3} \\ * (i \cos \alpha)^{0.7} z^{2/3} D_w^{1.8} = f_c (i \cos \alpha)^{0.7} z^{2/3} F(D_w) \quad (2-228)$$

The stress condition in every radial ball bearing is related to the bearing structure. Because of manufacturing error and other factors, the actual bearing basic dynamic load rating is little lower than the theoretical value. Therefore an empirical coefficient λ is introduced in bearing basic dynamic load rating equation, and its values are listed in Table 2-18.

Table 2 – 18 Ball bearing coefficient λ

Bearing type	λ
self-aligning ball bearing	1
single row radial ball bearing (without filling slot)	0.95
separable type single row angular contact ball bearing	0.95
single row and double row angular contact ball bearing	0.95
double row radial contact ball bearing	0.9
single row thrust ball bearing	0.9

Test result showed that when rolling element diameter is $D_w > 25.4\text{mm}$, ball bearing basic dynamic load rating is not proportional to D_w^{18} . So an empirical coefficient $F(D_w)$ is introduced.

When $D_w \leq 25.4\text{mm}$, $F(D_w) = D_w^{18}$.

When $D_w > 25.4\text{mm}$, $F(D_w) = 3.647 D_w^{14}$.

Putting λ and $F(D_w)$ into the radial bearing basic dynamic load rating equation we can get:

$$C = f_c (i \cos \alpha)^{0.7} z^{2/3} F(D_w) \tag{2-229}$$

Where
$$f_c = 39.9 \lambda \gamma^{0.3} \frac{(1-\gamma)^{1.39}}{(1+\gamma)^{1/3}} \left(\frac{2f_i}{2f_i-1} \right)^{0.41}$$
$$\left\{ 1 + \left[1.04 \left(\frac{1-\gamma}{1+\gamma} \right)^{1.72} \left(\frac{f_i}{f_e} \frac{(2f_e-1)}{(2f_i-1)} \right)^{0.41} \right]^{10/3} \right\}^{-0.3}$$
$$\gamma = \frac{D_w \cos \alpha}{D_{pw}}$$

For radial and angular contact ball bearing, when $f_i < 0.52$ and $f_e < 0.53$, because of the increase of friction, bearing load capacity can not be increased. So, when $f_i \leq 0.52$ and $f_e \leq 0.53$, $f_i = 0.52$ and $f_e = 0.53$ should be put into the above every equation. If $f_i > 0.52$ and $f_e > 0.53$, bearing basic dynamic load rating should be decreased and in calculating, the r_{imax} , r_{emax} , D_{wmin} should be put into the equation.

Radial and angular contact ball bearing f_c values are listed in Table 2 – 19. In Table 2 – 19, f_c values are suitable for

Radial and angular contact ball bearing:

$$f_i \leq 0.52, f_e \leq 0.53$$

Self-aligning ball bearing:

$$f_i \leq 0.53$$

Table 2 – 19 Radial and angular contact ball bearing f_c value

$\frac{D_w \cos \alpha}{D_{pw}}$	f_c			
	single row radial single double row angular contact ball bearing	double row radial ball bearing	self-aligning ball bearing	separable type single angular contact ball bearing
0.05	46.6	44.2	17.3	16.2
0.06	49.1	46.5	18.6	16.9
0.07	51.1	48.4	19.9	18.5
0.08	52.9	50.1	21.1	19.5
0.09	54.3	51.4	22.3	20.6
0.10	53.5	53.0	23.3	21.5
0.12	57.4	54.4	25.6	23.4
0.14	58.8	55.7	27.7	25.3
0.16	59.6	56.5	29.7	27.1
0.18	59.9	56.8	31.7	28.8
0.20	59.9	56.8	33.5	30.5
0.22	59.6	56.5	35.2	32.1
0.24	58.9	55.9	36.8	33.6
0.26	58.1	55.1	38.2	35.1
0.28	57.2	54.1	39.4	36.1
0.30	56.0	53.1	40.3	37.9
0.32	54.7	52.0	41.0	38.9
0.34	53.3	50.5	41.2	39.8
0.36	51.7	49.1	41.3	40.4
0.38	50.0	47.5	41.0	40.7
0.40	48.2	45.8	40.4	40.9

Note: 1. D_{pw} is the ball center circle diameter;

In the table, for the not given value of $D_w \cos \alpha / D_{pw}$, we can calculate from linear interpolation.

- When calculating the basic dynamic load rating of two identical single row radial ball bearings in pair arrangement, we can consider it as a double row radial ball bearing. When calculating the basic dynamic load rating of the back-to-back (outer ring wide face to face) or face-to-face (outer ring narrow face-to-face) pair arrangement of two identical single row angular contact radial ball bearings, we can consider it as a double row angular contact radial ball bearing. When calculating the basic dynamic load rating of the tandem pair arrangement (outer ring wide and narrow face-to-face) of two or more identical single row angular

contact radial ball bearings, if this group of bearings are precisely manufactured and correctly mounted letting load evenly distributed, their basic dynamic load rating can be the 0.7 power of bearing number times the value of a single row angular contact radial ball bearing basic dynamic load rating.

(2) Thrust and thrust radial ball bearing basic dynamic load rating

In rolling bearing subjecting to central axial load, every rolling element is subjecting to the same load. According to force equilibrium, we can get:

$$F_a = Qz \sin \alpha \quad (2-230)$$

In the equation (2-230), if Q designates the rolling element load rating Q_c , then F_a denotes the inner ring and outer ring load rating C_i , C_e . That is:

$$C_i = Q_{ci} z \sin \alpha \quad (2-231)$$

$$C_e = Q_{ce} z \sin \alpha \quad (2-232)$$

Besides

$$C = g_c C_i \quad (2-224)$$

Putting the equations (2-202) into equations (2-231), (2-232) and (2-224), and introducing for $\alpha = 90^\circ$:

$$\gamma = \frac{D_w}{D_{pw}} \quad (2-233)$$

1) while $\alpha = 90^\circ$:

$$C = f_c z^{2/3} F(D_w) \quad (2-234)$$

$$f_c = 98 \lambda \eta \gamma^{0.3} \left(\frac{2f_i}{2f_i - 1} \right)^{0.41} \left\{ 1 + \left[\left(\frac{f_i}{f_e} \frac{(2f_e - 1)}{(2f_i - 1)} \right)^{0.41} \right]^{10/3} \right\}^{-0.3} \quad (2-235)$$

2) While $\alpha \neq 90^\circ$:

$$\gamma = \frac{D_w \cos \alpha}{D_{pw}} \quad (2-17)$$

$$C = f_c (\cos \alpha)^{0.7} \tan \alpha z^{2/3} F(D_w) \quad (2-236)$$

$$f_c = 98 \lambda \eta \gamma^{0.3} \frac{(1 - \gamma)^{1.39}}{(1 + \gamma)^{1/3}} \left(\frac{2f_i}{2f_i - 1} \right)^{0.41} \left\{ 1 + \left[\left(\frac{(1 - \gamma)}{(1 + \gamma)} \right)^{1.72} \left(\frac{f_i (2f_e - 1)}{f_e (2f_i - 1)} \right)^{0.41} \right]^{10/3} \right\}^{-0.3} \quad (2-237)$$

In the equations (2-235) and (2-236):

When $D_w \leq 25.4 \text{ mm}$, $F(D_w) = D_w^{1.8}$.

When $D_w > 25.4 \text{ mm}$, $F(D_w) = 3.647 D_w^{1.4}$.

In the equations (2-235) and (2-237), λ value can be checked out from Table 2-18.

In thrust rolling bearing, because of manufacturing error, such as raceway wall thickness error etc., the actual bearing basic dynamic load rating is little lower than the theoretical value. Therefore two empirical coefficients λ and η are introduced in bearing basic dynamic load rating equation. The η value is related to contact angle:

$$\text{For ball bearing: } \eta = 1 - 1/3 \sin \alpha \quad (2-238)$$

$$\text{For roller bearing: } \eta = 1 - 0.15 \sin \alpha \quad (2-239)$$

(3) For single row or multiple row thrust ball bearing subjecting to single direction load and with the same ball diameter, its basic dynamic load rating can be calculated from the law of probability product.

If using $C_1, C_2, C_3 \dots C_n$ expresses every row basic dynamic load rating, $Z_1, Z_2, Z_3, \dots Z_n$ expresses every row rolling element number, and C expresses entire bearing basic dynamic load rating, then

$$Z = Z_1 + Z_2 + Z_3 + \dots + Z_n$$

From the equation (2-219) we can get:

$$\lg \frac{1}{S_1} = k_1 (Qz_1)^w L^e$$

$$\lg \frac{1}{S_2} = k_2 (Qz_2)^w L^e$$

$$\vdots$$

$$\lg \frac{1}{S_n} = k_n (Qz_n)^w L^e$$

But

$$\lg \frac{1}{S} = k (Qz)^w L^e$$

If taking

$$S_1 = S_2 = \dots = S = 0.9$$

$$L = 1$$

Then we can get

$$k_1 = \frac{\lg \frac{1}{0.9}}{C_1^w}$$

$$k_2 = \frac{\lg \frac{1}{0.9}}{C_2^w}$$

$$\vdots$$
$$k = \frac{\lg \frac{1}{0.9}}{C^w}$$

According to the law of probability product, we can get;

$$S = S_1 S_2 \dots S_n$$

Putting $k_1, k_2, \dots k$ values into every equation, we can get:

$$c = (z_1 + z_2 + \dots + z_n) \left[\left(\frac{z_1}{C_1} \right)^{10/3} + \left(\frac{z_2}{C_2} \right)^{10/3} + \dots + \left(\frac{z_n}{C_n} \right)^{10/3} \right]^{-0.3} \tag{2-240}$$

Thrust rolling bearing coefficient f_c values are listed in Table 2 – 20.

Table 2 – 20 Thrust rolling bearing coefficient f_c

$\frac{D_w}{D_{pw}}$	f_c	$\frac{D_w \cos \alpha}{D_{pw}}$	f_c		
	$\alpha = 90^\circ$		$\alpha = 45^\circ$	$\alpha = 60^\circ$	$\alpha = 75^\circ$
0. 01	36. 7	0. 01	42. 1	39. 1	37. 4
0. 02	45. 2	0. 02	51. 7	48. 1	45. 9
0. 03	51. 1	0. 03	58. 3	54. 2	51. 7
0. 04	55. 7	0. 04	63. 3	58. 8	56. 1
0. 05	59. 5	0. 05	67. 3	62. 7	59. 7
0. 06	62. 9	0. 06	70. 6	65. 7	62. 7
0. 07	65. 8	0. 07	73. 5	68. 4	65. 2
0. 08	68. 5	0. 08	75. 9	70. 0	67. 4
0. 09	71. 0	0. 09	77. 9	72. 6	69. 1
0. 10	73. 3	0. 10	79. 6	74. 1	70. 7
0. 12	77. 4	0. 12	82. 4	76. 7	
0. 14	81. 1	0. 14	84. 1	78. 3	
0. 16	84. 3	0. 16	85. 1	79. 2	
0. 18	87. 4	0. 18	85. 5	79. 6	
0. 20	90. 2	0. 20	85. 4	79. 5	
0. 22	92. 9	0. 22	84. 9		
0. 24	95. 5	0. 24	83. 9		
0. 26	97. 6	0. 26	82. 8		
0. 28	100. 0	0. 28	81. 3		
0. 30	102. 0	0. 30	79. 5		
0. 32	104. 0				
0. 34	105. 9				

Values listed in Table 2 – 20 are suitable for thrust ball bearing of $f_i = f_e \leq 0.54$.

If groove curvature radius is larger than $0.54D_w$, we should calculate f_e value from the equations (2 – 235) or (2 – 237).

4. Roller bearing basic dynamic load rating

(1) Basic dynamic load rating theoretical equation for line contact roller bearing

There are many roller bearing contacting types. When determining basic dynamic load rating, it should be considered that contacting type is an influential factor, but also manufacturing and mounting error.

For rolling element contacting with both two ring raceways in line contact, basic dynamic load rating theoretical equation can be determined by the following method:

Radial roller bearing:

Putting the equation (2 – 203) into the equations (2 – 217) and (2 – 218), we can calculate C_i and C_e , then from the equation (2 – 224) we can get:

$$C = 207.9\gamma^{\frac{2}{9}} \frac{(1-\gamma)^{\frac{29}{27}}}{(1+\gamma)^{\frac{1}{4}}} \left\{ 1 + \left[1.04 \left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{108}} \right]^{\frac{9}{2}} \right\}^{-\frac{2}{9}} \\ * (il_{we} \cos \alpha)^{\frac{7}{9}} D_w^{\frac{29}{27}} z^{\frac{3}{4}} \quad (2-241) \\ \gamma = \frac{D_w \cos \alpha}{D_{pw}}$$

Thrust roller bearing:

Under central axial load, every rolling element is subjected to the same load. From force equilibrium, we can get:

$$F_a = zQ \sin \alpha$$

If in the above equation, Q expresses the basic rolling element load rating Q_e , then F_a expresses the basic load rating C_i and C_e of inner ring and outer ring. And

$$C_i = zQ_{ci} \sin \alpha \quad (2-231)$$

$$C_e = zQ_{ce} \sin \alpha \quad (2-232)$$

Putting the equation (2 – 203) into the equations (2 – 231), (2 –

232) and (2-224), we can get:

$$C = 551 \gamma^{\frac{2}{9}} \frac{(1-\gamma)^{\frac{29}{27}}}{(1+\gamma)^{\frac{1}{4}}} \left[1 + \left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{24}} \right]^{-\frac{2}{9}} \\ * (l_{we} \cos \alpha)^{\frac{7}{9}} \tan \alpha \cdot D_w^{\frac{29}{27}} z^{\frac{3}{4}} \quad (2-242)$$

For $\alpha = 90^\circ$ thrust roller bearing, if putting $\gamma = D_w \cos \alpha / D_{pw}$ into the equation (2-242), due to:

$$\cos^{\frac{2}{9}} \alpha \cos^{\frac{7}{9}} \alpha \tan \alpha = 1$$

Therefore, $\alpha = 90^\circ$ thrust roller bearing basic dynamic load rating calculating equation is:

$$C = 551 * 2^{-\frac{2}{9}} \left(\frac{D_w}{D_{pw}} \right)^{\frac{2}{9}} l_{we}^{\frac{7}{9}} z^{\frac{3}{4}} D_w^{\frac{29}{27}} \quad (2-243)$$

(2) Roller bearing contact types

When the generatrices of roller and raceway are straight lines this case is called as line contact. When the generatrix of roller is a curve line (convex roller) or the raceway generatrix is a curve line (convex raceway), the contact type related to roller load, and can not be decided easily. For a given convex value, there is an optimum rolling element load. Under the rolling element load, the rolling element is subjecting to the load evenly distributed along its length direction, and in this case, there is no edging stress. If the actual rolling element load is larger than the given optimum rolling element load, there will be edging stress, and in this case the bearing life will be lowered. If the actual rolling element load is lower than the given optimum rolling element load, there will be no edging stress and some portion of the rollig element in length direction will not subject to load, and in this case the bearing load capacity would not be completely used.

Knowing from bearing load distribution analysis, the rolling element load is related with the circumference position where rolling element is located. Therefore, for a given convex value, the contact type for rolling element-raceway is different with different rolling element locating circumference position during a bearing running circle. Even in a bearing, the contact types of rolling element with inner ring raceway and outer ring

raceway also are different.

For a given bearing application, it is the optimum design plan that under a given applied load, there is no edging stress existing on the contact position where the maximum rolling element load contacts with raceway. For a general bearing application, it is the first step to assume a ratio of applied bearing load to basic dynamic load rating, and then to determine the convex value.

Usually, contact type for roller-raceway can be defined by the relationship of the major axis ($2a$) of contact ellipse with the effective roller length.

1) Line contact

Line contact designates the contact that when there is no applied load and mounting condition is good with better concentricity, roller contacts raceway with their full generatrix length; or that under an applied load, the major axis of contact ellipse is $2a > 1.5 l_{we}$ (l_{we} is the effective roller length).

2) Point contact

Point contact designates the contact that when there is no applied load and mounting condition is good with better concentricity, roller contacts with raceway on its middle point; or that under an applied load, the major axis of the contact ellipse where the maximum rolling element load contacts with raceway is smaller than the effective roller length, that is $2a < l_{we}$.

3) Modified line contact

Modified line contact designates the contact that modifying the generatrix shape at two ends of roller or raceway makes the contact stress at the ends of the roller subjecting to the maximum rolling element load close to the contact stress at other portions of the roller, or makes the major axis of the contact ellipse in $l_{we} < 2a < 1.5 l_{we}$ range.

Several contact types and corresponding conditions that may exist in roller bearings are listed in Table 2-21.

Table 2-21 Roller bearing contact type

Contact type		Condition	
inner ring	outer ring	$2a_i$	$2a_e$
line contact	line contact	$2a_i > 1.5 l_{we}$	$2a_e > 1.5 l_{we}$

(continued)

Contact type		Condition	
inner ring	outer ring	$2a_i$	$2a_e$
point contact	point contact	$2a_i < 1.5 l_{we}$	$2a_e < 1.5 l_{we}$
line contact	point contact	$2a_i > 1.5 l_{we}$	$2a_e < l_{we}$
point contact	line contact	$2a_i < l_{we}$	$2a_e > 1.5 l_{we}$
modified contact	modified contact	$l_{we} \leq 2a_i \leq 1.5 l_{we}$	$l_{we} \leq 2a_e \leq 1.5 l_{we}$

(3) Reducing coefficient λ , η

A large number of test data have proved that the actual roller bearing load capacity is lower than the theoretical calculating value of dynamic load rating. So a reducing coefficient λ should be introduced to the theoretical calculating equation. The value of the reducing coefficient λ is related to the bearing structure type , the roller landing mode and the contact type etc. The reducing coefficient λ for every type bearing is listed in Table 2 - 22.

In thrust roller bearing, because of manufacturing error and mounting error etc. a reducing coefficient η also need to be introduced.

$$\eta = 1 - 0.15\sin\alpha$$

(2 - 239)

Table 2 - 22 λ , ν value

Bearing type	λ	ν	$\lambda\nu$
short cylindrical roller bearing and tapered roller bearing that both ring contacts are line contacts	0.45	1.36	0.61
roller bearing that one ring contact is point contact, the other ring contact is line contact; aligning roller bearing with fixed rib; cylindrical roller bearing with one raceway with crown; tapered roller bearing with one raceway with crown; thrust aligning roller bearing	0.54	1.26	0.68
roller bearing that both ring contacts are modified line contacts, and roller length is $l \leq 2.5D_w$	0.61	1.36	0.83 ^①
aligning roller bearing that both ring contacts are modified line contacts with loose center ring; tapered roller bearing that both ring contacts are modified line contacts.	0.65	1.2	0.78 ^②

Note: ① This value originally stipulated by ISO/R281 corresponded to $\lambda\nu = 0.73$. In 1971, TC4 decided to enhance the value corresponding to $\lambda\nu = 0.83$.

② ISO/R281 only stipulated the f_c/f value for roller bearings. In Palmgren book (1964) this value for aligning roller bearing was given independently.

(4) Modifying coefficient V in the equation (2-188)

Knowing from Table 2-16, the life exponent is $\varepsilon = 3$ for point contact; $\varepsilon = 4$ for line contact.

There are many kinds of structure shapes for roller bearings. But the contact types of the roller bearings are very complicated. Some are point contact, and some are line contact. Even in one bearing, rolling element can contact with one ring raceway for point contact, but can contact with the other ring raceway for line contact. For the same ring, its contact type can be changed from point contact to line contact within different load range. It is obvious that the life exponent ε changes from 3 ~ 4 for roller bearings.

For simplifying roller bearing life calculation, it is better to take one uniform value of $\varepsilon = 10/3$ for roller bearing life exponent. Then a modifying coefficient ν is introduced into the theoretical calculation equation (2-241) and (2-242) for line contact roller bearing basic dynamic load rating. If we assume that C_1 expresses the theoretical value of line contact roller bearing basic dynamic load rating from the theoretical calculation equation (2-241) and (2-242), then we can express the theoretical value of roller bearing basic dynamic load rating by the following equation for all contact types:

$$C = \nu C_1 \quad (2-244)$$

When determining the coefficient ν , bearing life should be in commonly used scope, that is within 100 (million rotations) $< L < 10000$ (million rotations), and the error is minimum in this range.

In the equation (2-188), the Q_c is directly proportional with the basic dynamic load rating C . So under the same applied load, we can get:

$$L = \left(\frac{C}{P} \right)^\varepsilon \quad (2-271)$$

Where P is the dynamic equivalent load. See Section 2.6.4 of this chapter.

For roller contacts are line contacts with both rings, the life exponent should be 4. Then the above equation can be written as:

$$C_1 = L^{\frac{1}{4}} P \quad (2-245)$$

If changing life exponent as $10/3$, then we can get:

$$C = L^{\frac{3}{10}} P \quad (2-246)$$

From the equations (2-244), (2-245) and (2-246) we can get:

$$\nu = L^{\frac{3}{10} - \frac{1}{4}} \quad (2-247)$$

If taking $L = 500$ (million rotations)

Then we can get:

$$\nu = 500^{\frac{3}{10} - \frac{1}{4}} = 1.36$$

If roller contacts are point contacts with both rings, then under certain applied load, there will be changing from point contact to line contact. Assume that this change occurs at $L = 100$ (million rotations), and both rings changes at the same time. The life exponent ε will be changing from 3 to 4. In this condition, if using $\varepsilon = 10/3$, there will be 5% error. From the equation (2-247) we can get:

$$\nu = 0.95 * 100^{\frac{3}{10} - \frac{1}{4}} = 1.20$$

If roller contact is point contact with one ring, with the other ring is line contact, Assume that this change from point contact to line contact occurs at $L = 100$ (million rotations). From the equation (2-247) we can get:

$$\nu = 100^{\frac{3}{10} - \frac{1}{4}} = 1.26$$

The ν values of every condition are listed in Table 2-22.

(5) Roller bearing basic dynamic load rating calculation equation

If introducing λ , η , ν into the theoretical calculation equation of line contact roller bearing basic dynamic load rating, then we can get the general expression of roller bearing basic dynamic load rating.

Radial roller bearing:

$$C = f_c (i L_{we} \cos \alpha)^{\frac{7}{9}} z^{\frac{3}{4}} D_w^{\frac{29}{27}} \quad (2-248)$$

$$f_c = 207.9 \lambda \nu \gamma^{\frac{2}{9}} \frac{(1-\gamma)^{\frac{29}{27}}}{(1+\gamma)^{\frac{1}{4}}} \left\{ 1 + \left[1.04 \left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{108}} \right]^{\frac{9}{2}} \right\}^{-\frac{2}{9}} \quad (2-249)$$

$$\gamma = \frac{D_w \cos \alpha}{D_{pw}}$$

Thrust radial roller bearing: $\alpha \neq 90^\circ$

$$C = f_c (l_{we} \cos \alpha)^{\frac{7}{9}} \tan \alpha D_w^{\frac{29}{27}} z^{\frac{3}{4}} \quad (2-250)$$

$$f_c = 551 \lambda \eta \nu \gamma^{\frac{2}{9}} \frac{(1-\gamma)^{\frac{29}{27}}}{(1+\gamma)^{\frac{1}{4}}} \left[1 + \left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{24}} \right]^{-\frac{2}{9}} \quad (2-251)$$

Thrust roller bearing: $\alpha = 90^\circ$

$$C = f_c l_{we}^{\frac{7}{9}} z^{\frac{3}{4}} D_w^{\frac{29}{27}} \quad (2-252)$$

$$f_c = 551 * 2^{-\frac{2}{9}} \lambda \eta \nu \left(\frac{D_w}{D_{pw}} \right)^{\frac{2}{9}} \quad (2-253)$$

Multiple row single direction thrust roller bearing:

For multiple row or multiple row thrust roller bearing subjected to single direction axial load, we can use the same deducing process as the equation (2-240) to decide the basic dynamic load rating equation.

If $Z_1, Z_2, Z_3, \dots, Z_n$ expresses every row rolling element number, $l_{we1}, l_{we2}, l_{we3} \dots l_{wen}$ expresses every row effective roller length, and $C_1, C_2, \dots C_n$ expresses every row bearing basic dynamic load rating, then the basic dynamic load rating for the entire bearing can be calculated from the following equation:

$$C = (z_1 l_{we1} + z_2 l_{we2} + \dots + z_n l_{wen}) \left[\left(\frac{z_1 l_{we1}}{C_1} \right)^{\frac{9}{2}} + \left(\frac{z_2 l_{we2}}{C_2} \right)^{\frac{9}{2}} + \dots + \left(\frac{z_n l_{wen}}{C_n} \right)^{\frac{9}{2}} \right]^{-\frac{2}{9}} \quad (2-254)$$

The f_c values for radial roller bearing and thrust roller bearing are listed in Table 2-23 and Table 2-24.

Table 2-23 Radial roller bearing f_c

$\frac{D_w \cos \alpha}{D_{pw}}$	f_c			
	$\lambda \nu = 0.61$	$\lambda \nu = 0.68$	$\lambda \nu = 0.73$	$\lambda \nu = 0.83$
0.01	38.2	42.7	45.8	52.1
0.02	44.6	49.7	53.4	60.7
0.03	48.8	54.4	58.4	66.4
0.04	52.0	57.9	62.2	70.7
0.05	54.4	60.7	65.1	74.0
0.06	56.5	63.0	67.6	76.9
0.07	58.2	64.9	69.6	79.1
0.08	59.6	66.5	71.4	81.2
0.09	60.9	67.9	72.9	82.9

(continued)

$\frac{D_w \cos \alpha}{D_{pw}}$	f_c			
	$\lambda \nu = 0.61$	$\lambda \nu = 0.68$	$\lambda \nu = 0.73$	$\lambda \nu = 0.83$
0.10	61.9	69.0	74.0	84.2
0.11	62.7	69.9	75.1	85.3
0.12	63.5	70.7	75.9	86.3
0.13	63.9	71.4	76.6	87.1
0.14	64.4	71.9	77.2	86.7
0.15	64.8	72.3	77.6	88.2
0.16	65.0	72.5	77.9	88.5
0.17	65.2	72.7	78.1	88.8
0.18	65.2	72.8	78.1	88.9
0.19	65.2	72.8	78.1	88.8
0.20	65.1	72.7	78.0	88.7
0.21	65.0	72.5	77.9	88.5
0.22	64.8	72.3	77.6	88.2
0.23	64.5	72.0	77.3	87.9
0.24	64.2	71.6	76.9	87.5
0.25	63.8	71.2	76.5	87.0
0.26	63.5	70.8	76.0	86.4
0.27	63.1	70.3	75.5	85.8
0.28	62.6	69.7	74.9	85.1
0.29	62.1	69.2	74.3	84.4
0.30	61.5	68.5	73.6	83.7

Table 2 – 24 Thrust roller bearing f_c

$\frac{D_w}{D_{pw}}$	f_c $\alpha = 90^\circ$	$\frac{D_w \cos \alpha}{D_{pw}}$	f_c		
			$\alpha = 50^\circ \textcircled{1}$	$\alpha = 65^\circ \textcircled{2}$	$\alpha = 80^\circ \textcircled{3}$
0.01	105.4	0.01	109.0	107.0	105.6
0.02	123.0	0.02	127.9	124.7	123.1
0.03	134.5	0.03	139.5	136.2	134.4
0.04	143.5	0.04	148.4	144.7	142.9
0.05	150.7	0.05	155.2	151.5	149.5
0.06	157.0	0.06	160.9	157.0	154.9
0.07	162.4	0.07	165.6	161.6	159.5
0.08	167.2	0.08	169.6	165.5	163.3
0.09	171.7	0.09	172.9	168.8	165.4

(continued)

$\frac{D_w}{D_{pw}}$	f_c $\alpha = 90^\circ$	$\frac{D_w \cos \alpha}{D_{pw}}$	f_c		
			$\alpha = 50^\circ$ ①	$\alpha = 65^\circ$ ②	$\alpha = 80^\circ$ ③
0.10	175.7	0.10	175.5	171.4	169.1
0.12	183.1	0.12	179.8	175.4	173.1
0.14	189.5	0.14	182.3	178.0	175.5
0.16	195.2	0.16	183.8	179.4	
0.18	200.3	0.18	184.2	178.8	
0.20	205.1	0.20	183.8	179.4	
0.22	207.5	0.22	182.7		
0.24	213.6	0.24	180.9		
0.26	217.3	0.26	178.8		
0.28	220.9				
0.30	224.4				

Note: ① Suitable for $45^\circ < \alpha < 60^\circ$;

② Suitable for $60^\circ < \alpha < 75^\circ$;

③ Suitable for $75^\circ < \alpha < 90^\circ$.

The value that is not listed in the table can be calculated from linear interpolation.

The listed values correspond to $\lambda\nu = 0.73$. ISO/R281 recommended the values corresponding to $\lambda\nu = 0.68$. Later in 1971 deciding to use this table values.

In Table 2-23, the f_c values corresponding to $\lambda\nu = 0.83$ column are suitable for:

1) For short cylindrical roller bearing, the roller contacts with both rings in modified line contacts, the roller is precisely guided by the cage, and two ribs of any ring precisely guide its faces.

2) For tapered roller bearing, the roller contacts with both rings in modified line contacts, and the roller is precisely guided by one rib.

3) For aligning roller bearing, the roller contacts with both rings in modified line contacts, and the symmetrical spherical roller is precisely guided by axial loose ring.

For other radial roller bearings which condition is different with the above conditions, the $\lambda\nu$ value can be checked out from Table 2-22, and then the f_c value can be calculated from the equation (2-251) or be checked out from Table 2-23.

ISO 281/I-1977 recommended to take a uniform f_c value for cylindrical roller bearing, tapered roller bearing, and aligning roller bearing. The listing values corresponding to $\lambda\nu = 0.83$ column in Table 2 – 23 can be considered as the maximum values for these three kinds of roller bearings.

Table 2 – 24 lists the maximum value that is only suitable for the contact stress evenly distributed over the contact position where the maximum load roller contacts with raceway under applied load. If there exists stress concentration on some portions of roller-raceway contact under applied load, it is better to take a lower value than the table listed value. Such as, roller bearing in which there is existing edging stress; bearing with not precisely guided roller; bearing that its roller length is larger than 2.5 times of roller diameter; bearing that its roller length is too long causing more sliding. For these bearings, it is better to take a lower value.

The f_c values for every type bearing are listed in Table 2 – 25.

2.6.4 Equivalent Dynamic Load of Rolling Bearing

Knowing from the aforementioned that rolling bearing basic dynamic load rating is determined under assumed running conditions. That is the radial load for radial bearing assuming that the inner ring is rotating and the outer ring is in stationary; for thrust bearing, that is the central axial load; for angular contact bearing, that is the radial component of a load under which the bearing is just half-circle raceway loading.

If the actual applied load is different from the assumed condition, then the actual applied load has to be changed to the assumed load under the same assumed running condition of determining the basic dynamic load rating. Under the assumed load, the bearing life is as the same as the life of the actual applied load.

Therefore, the assumed load is called as the dynamic equivalent load P .

1. Rolling bearing rating life calculation equation

The relationship of the rating life with the rolling element rating load and the actual rolling element load has been given in the equation (2 – 188):

Table 2 – 25 Rolling bearing basic dynamic load rating calculation equation

Bearing type		C	f_c	γ
radial and angular contact ball bearing		$f_c (i \cos \alpha)^{0.7} z^{\frac{2}{3}} F(D_w)$	$39.9 \lambda \gamma^{0.3} \frac{(1-\gamma)^{1.39}}{(1+\gamma)^{\frac{1}{3}}} \left(\frac{2f_i}{2f_i-1} \right)^{0.41}$ $\left\{ 1 + \left[1.04 \left(\frac{1-\gamma}{1+\gamma} \right)^{1.72} \left(\frac{f_i(2f_e-1)}{f_e(2f_i-1)} \right)^{0.41} \right]^{\frac{10}{3}} \right\}^{-0.3}$	$\frac{D_w \cos \alpha}{D_{pw}}$
thrust and thrust angular radial ball bearing	$\alpha = 90^\circ$	$f_c z^{\frac{2}{3}} F(D_w)$	$98 \lambda \eta \gamma^{0.3} \left(\frac{2f_i}{2f_i-1} \right)^{0.41} \left\{ 1 + \left[\left(\frac{f_i(2f_e-1)}{f_e(2f_i-1)} \right)^{0.41} \right]^{\frac{10}{3}} \right\}^{-0.3}$	$\frac{D_w}{D_{pw}}$
	$\alpha \neq 90^\circ$	$f_c (\cos \alpha)^{0.7} \tan \alpha z^{\frac{2}{3}} F(D_w)$	$98 \lambda \eta \gamma^{0.3} \frac{(1-\gamma)^{1.39}}{(1+\gamma)^{1/3}} \left(\frac{2f_i}{2f_i-1} \right)^{0.41}$ $\left\{ 1 + \left[\left(\frac{1-\gamma}{1+\gamma} \right)^{1.72} \left(\frac{f_i(2f_e-1)}{f_e(2f_i-1)} \right)^{0.41} \right]^{\frac{10}{3}} \right\}^{-0.3}$	$\frac{D_w \cos \alpha}{D_{pw}}$
	multiple row thrust ball bearing	$C = (z_1 + z_2 + \dots + z_n) \left[\left(\frac{z_1}{C_1} \right)^{\frac{10}{3}} + \left(\frac{z_2}{C_2} \right)^{\frac{10}{3}} + \dots + \left(\frac{z_n}{C_n} \right)^{\frac{10}{3}} \right]^{-0.3}$		
radial cylindrical roller bearing cylindrical roller bearing tapered roller bearing aligning roller bearing		$f_c (i l_w \cos \alpha)^{\frac{7}{9}} z^{\frac{3}{4}} D_w^{\frac{29}{27}}$	$207.9 \lambda \nu \gamma^{\frac{2}{9}} \frac{(1-\gamma)^{\frac{29}{27}}}{(1+\gamma)^{1/4}} \left\{ 1 + \left[1.04 \left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{108}} \right]^{\frac{9}{2}} \right\}^{-\frac{2}{9}}$	$\frac{D_w \cos \alpha}{D_{pw}}$

(continued)

Bearing type		C	f_c	γ
thrust and thrust	$\alpha = 90^\circ$	$f_c l_{we} \frac{7}{9} z^{\frac{3}{4}} D_w^{\frac{29}{27}}$	$551 * 2^{-\frac{2}{9}} \lambda \eta \nu \gamma^{\frac{2}{9}}$	$\frac{D_w}{D_{pw}}$
angular contact roller bearing	$\alpha \neq 90^\circ$	$f_c (l_{we} \cos \alpha)^{\frac{7}{9}} \tan \alpha z^{\frac{3}{4}} D_w^{\frac{29}{27}}$	$551 \lambda \eta \nu \gamma^{\frac{2}{9}} \frac{(1-\gamma)^{\frac{29}{27}}}{(1+\gamma)^{1/4}} \left[1 + \left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{24}} \right]^{-\frac{2}{9}}$	$\frac{D_w \cos \alpha}{D_{pw}}$
multiple row bearing		$C = (z_1 l_{y1} + z_2 l_{y2} + \dots + z_n l_{yn}) \left[\left(\frac{z_1 l_{y1}}{C_1} \right)^{\frac{9}{2}} + \left(\frac{z_2 l_{y2}}{C_2} \right)^{\frac{9}{2}} + \dots + \left(\frac{z_n l_{yn}}{C_n} \right)^{\frac{9}{2}} \right]^{-\frac{2}{9}}$		

Note: 1. When $D_w \leq 25.4 \text{ mm}$, $F(D_w) = D_w^{1.8}$.

When $D_w > 25.4 \text{ mm}$, $F(D_w) = 3.647 D_w^{1.4}$.

2. For ball bearing, $\eta = 1 - \frac{1}{3} \sin \alpha$.

For roller bearing, $\eta = 1 - 0.15 \sin \alpha$.

3. The λ , ν values can be checked out from Table 2-18 and 2-22.

Symbols:

C — bearing basic dynamic load rating (N);

α — contact angle;

z — one row rolling element number;

i — rolling element rows;

D_w — rolling element diameter (mm);

L_{we} — rolling element effective contact length (mm).

$$L = \left(\frac{Q_c}{Q} \right)^\varepsilon \quad (2-188)$$

Where Q_c — rolling element rating load;

Q — actual rolling element load;

ε — life exponent.

If using L_i and L_e express the rating life of rolling element contacting with inner ring and with outer ring respectively, then we can get:

$$L_i = \left(\frac{Q_{ci}}{Q_i} \right)^\varepsilon \quad (2-255)$$

$$L_e = \left(\frac{Q_{ce}}{Q_e} \right)^\varepsilon \quad (2-256)$$

Where Q_i and Q_e are the even rolling element load at the place where the inner ring and the outer ring contact respectively. From the equations (2-211), (2-212) and load distribution analysis we can get:

$$Q_i = \frac{F_r}{z \cos \alpha} \frac{J_1(T)}{J_r(T)} = \frac{F_a}{z \sin \alpha} \frac{J_1(T)}{J_a(T)} \quad (2-257)$$

$$Q_e = \frac{F_r}{z \cos \alpha} \frac{J_2(T)}{J_r(T)} = \frac{F_a}{z \sin \alpha} \frac{J_2(T)}{J_a(T)} \quad (2-258)$$

From the equations (2-213) and (2-214) we can get:

$$Q_{ci} = \frac{C_i}{z \cos \alpha} \frac{J_1(0.5)}{J_r(0.5)} \quad (2-259)$$

$$Q_{ce} = \frac{C_e}{z \cos \alpha} \frac{J_2(0.5)}{J_r(0.5)} \quad (2-260)$$

Putting the equations (2-257) ~ (2-260) into the equations (2-255) and (2-256), we can get:

$$\begin{aligned} L_i &= \left[\frac{C_i}{F_r} \frac{J_1(0.5)}{J_r(0.5)} \frac{J_r(T)}{J_1(T)} \right]^\varepsilon \\ &= \left[\frac{C_i}{F_a} \frac{J_1(0.5)}{J_r(0.5)} \frac{J_a(T)}{J_1(T)} \tan \alpha \right]^\varepsilon \\ &= \left(\frac{C_i}{P_i} \right)^\varepsilon \end{aligned} \quad (2-261)$$

$$L_e = \left[\frac{C_e}{F_r} \frac{J_2(0.5)}{J_r(0.5)} \frac{J_r(T)}{J_2(T)} \right]^\varepsilon$$

$$\begin{aligned}
&= \left[\frac{C_e}{F_a} \frac{J_2}{J_r} \frac{(0.5)}{(0.5)} \frac{J_a}{J_2} \frac{(T)}{(T)} \tan \alpha \right]^e \\
&= \left(\frac{C_e}{P_e} \right)^e
\end{aligned} \tag{2-262}$$

From the equations (2-261) and (2-262) we can get:

$$\begin{aligned}
P_i &= F_r \frac{J_r}{J_1} \frac{(0.5)}{(0.5)} \frac{J_1}{J_r} \frac{(T)}{(T)} \\
&= F_a \frac{J_r}{J_1} \frac{(0.5)}{(0.5)} \frac{J_1}{J_a} \frac{(T)}{(T)} \cot \alpha
\end{aligned} \tag{2-263}$$

$$\begin{aligned}
P_e &= F_r \frac{J_r}{J_2} \frac{(0.5)}{(0.5)} \frac{J_2}{J_r} \frac{(T)}{(T)} \\
&= F_a \frac{J_r}{J_2} \frac{(0.5)}{(0.5)} \frac{J_2}{J_a} \frac{(T)}{(T)} \cot \alpha
\end{aligned} \tag{2-264}$$

P_i and P_e are called as inner ring and outer ring dynamic equivalent load respectively.

In the equations (2-220) and (2-221), if F_r is the inner ring and outer ring dynamic equivalent load P_i and P_e respectively, we can get:

$$\lg \frac{1}{S_i} = \frac{\lg \frac{1}{0.9}}{C_i^w} P_i^w L_i^e = \left(\frac{P_i}{C_i} \right)^w L_i^e \lg \frac{1}{0.9} \tag{2-265}$$

$$\lg \frac{1}{S_e} = \left(\frac{P_e}{C_e} \right)^w L_e^e \lg \frac{1}{0.9} \tag{2-266}$$

From the Law of Probability Product we can get:

$$\begin{aligned}
\lg \frac{1}{S} &= \lg \frac{1}{S_i S_e} = \left(\frac{P_i}{C_i} \right)^w L_i^e \lg \frac{1}{0.9} + \left(\frac{P_e}{C_e} \right)^w L_e^e \lg \frac{1}{0.9} \\
&= \left(\frac{P}{C} \right)^w L^e \lg \frac{1}{0.9}
\end{aligned} \tag{2-267}$$

For a bearing: $L_i = L_e = L$

Therefore we can get:

$$\left(\frac{P}{C} \right)^w = \left(\frac{P_i}{C_i} \right)^w + \left(\frac{P_e}{C_e} \right)^w \tag{2-268}$$

Putting the P_i and P_e equations (2-263) and (2-264) into the equation

(2-268), we can get:

$$P = F_r \frac{J_r(0.5)}{J_r(T)} \left[\left(\frac{C}{C_i} \frac{J_1(T)}{J_1(0.5)} \right)^w + \left(\frac{C}{C_c} \frac{J_2(T)}{J_2(0.5)} \right)^w \right]^{\frac{1}{w}} \quad (2-269)$$

Or

$$P = F_a \cot \alpha \frac{J_r(0.5)}{J_a(T)} \left[\left(\frac{C}{C_i} \frac{J_1(T)}{J_1(0.5)} \right)^w + \left(\frac{C}{C_c} \frac{J_2(T)}{J_2(0.5)} \right)^w \right]^{\frac{1}{w}} \quad (2-270)$$

From the equation (2-267) we can get the rolling bearing rating life calculation equation as:

$$L = \left(\frac{C}{P} \right)^{\frac{w}{\varepsilon}} = \left(\frac{C}{P} \right)^{\varepsilon} \quad (2-271)$$

Where L — rating life (million rotations);

C — basic dynamic load rating (N);

P — dynamic equivalent load (N);

ε ^① — life exponent.

For ball bearing: $\varepsilon = 3$

For roller bearing: $\varepsilon = 10/3$

The equation (2-271) is the basic equation for rolling bearing life calculation.

The equation (2-269) or (2-270) is the basic equation for rolling bearing dynamic equivalent load calculation. Knowing from the equation (2-269) or (2-270), the dynamic equivalent load is a function of radial

Note: ① In the equation (2-271), $\varepsilon = w/e$, but $w = 10/3$, $e = 10/9$ for point contact, so takes $\varepsilon = 3$.

For line point contact, $w = 9/2$, $e = 9/8$, so takes $\varepsilon = 4$.

Because there are many roller bearing contact types, such as point contact and line contact, in order to unify roller bearing life calculation method, we take $\varepsilon = 10/3$. In order to compensate the life exponent changing, a modifying coefficient v is introduced in the basic dynamic load rating calculation equation. See Section 2.6.3 of this chapter.

load F_r and axial load F_a . For a given rolling bearing type, first we determine the load distribution factor T from radial load F_r and axial load F_a . Then we can calculate out the rolling bearing dynamic equivalent load P corresponding to radial load F_r and axial load F_a by using the equation (2-269) or (2-270),

It is very inconvenient to use this very complicated calculation process. Next we will discuss a convenient and easy to use calculation equation for rolling bearing dynamic equivalent load calculation under a certain simplified condition.

2. Single row bearing dynamic equivalent load

From the equations (2-269) and (2-270) we can get:

$$\frac{F_r}{P} = \left[\left(\frac{C}{C_i} \frac{J_1(T)}{J_1(0.5)} \right)^w + \left(\frac{C}{C_e} \frac{J_2(T)}{J_2(0.5)} \right)^w \right]^{-\frac{1}{w}} \frac{J_r(T)}{J_r(0.5)} \quad (2-272)$$

$$\begin{aligned} \frac{F_a \cot \alpha}{P} &= \left[\left(\frac{C}{C_i} \frac{J_1(T)}{J_1(0.5)} \right)^w + \left(\frac{C}{C_e} \frac{J_2(T)}{J_2(0.5)} \right)^w \right]^{-\frac{1}{w}} \\ &\quad \frac{J_a(T)}{J_r(0.5)} \end{aligned} \quad (2-273)$$

From the equations (1-272) and (2-273) we can see that F_r/P and $F_a \cot \alpha / P$ are only related to the load distribution factor T . Figure 2-43 and Figure 2-44 show the relationships of F_r/P and $F_a \cot \alpha / P$ for point contact and line contact, respectively. In these figures, two curves express the extreme condition of $C_i/C_e \rightarrow 0$ and $C_e/C_i \rightarrow 0$.

Figure 2-43 and Figure 2-44 also show the relationships of radial load F_r and axial load F_a for getting the same dynamic equivalent load P .

In order to simplify the dynamic equivalent load P calculation, in these figures we use straight lines AB and BC instead of curve lines. The point of intersection of line AB and line BC is:

$$\left. \begin{aligned} \frac{F_r}{P} &= 1 \\ \frac{F_a}{P} &= e \end{aligned} \right\}$$

So, we can get the relationship of F_r and F_a at point B as:

$$\frac{F_a}{F_r} = e \quad e = \xi \tan \alpha$$

Where ξ is a constant factor related to contact type, and e is a factor related to ξ and contact angle α .

When $F_a/F_r \leq e$, we should calculate the dynamic equivalent load P by using the equation of line AB.

When $F_a/F_r > e$, we should calculate the dynamic equivalent load P by using the equation of line BC.

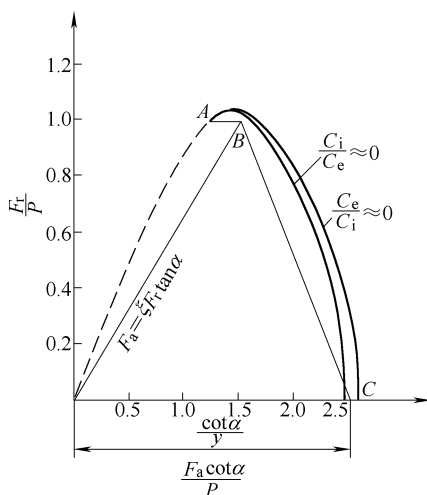


Figure 2-43 F_r and F_a relationship for point contact

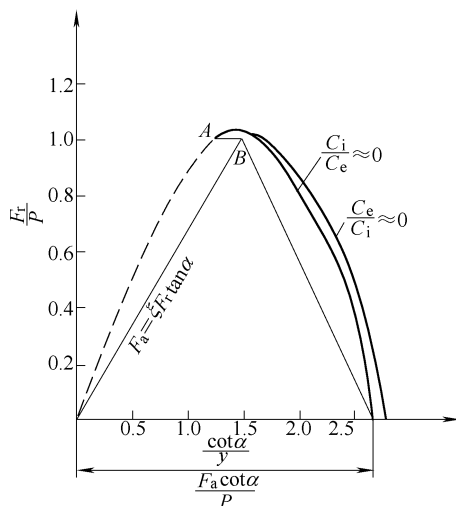


Figure 2-44 F_r and F_a relationship for line contact

When the relative value of F_r and F_a is smaller than the value of A point, the rolling element number subjecting to load is decreased so that the bearing can not work normally. Therefore the left part of the curve is expressed by dotted line.

Dynamic equivalent load P can be calculated by using a uniform equation:

$$P = XF_r + YF_a \quad (2-274)$$

For line AB : $X = 1$ $Y = 0$

For line BC : we can calculate it from the coordinates of point B and point C :

The coordinate of point C is as:

$$\frac{F_r}{P} = 0$$

$$\frac{F_a \cot \alpha}{P} = \left[\left(\frac{C}{C_i} \cdot \frac{1}{J_1(0.5)} \right)^w + \left(\frac{C}{C_e} \cdot \frac{1}{J_2(0.5)} \right)^w \right]^{-\frac{1}{w}} \\ * \frac{1}{J_r(0.5)}$$

Putting into the equation (2-274) we can get:

$$Y = A \cot \alpha \quad (2-275)$$

$$A = \left[\frac{1 + \left(\frac{C_i}{C_e} \cdot \frac{J_1(0.5)}{J_2(0.5)} \right)^w}{1 + \left(\frac{C_i}{C_e} \right)^w} \right]^{\frac{1}{w}} \frac{J_r(0.5)}{J_1(0.5)} \quad (2-276)$$

For different C_i/C_e value, we can calculate out the A value. But it has been found that the changing range of A value is very small, therefore, we can take the uniform value for point and line contacts as:

$$A \approx 0.4$$

$$\text{So} \quad Y = 0.4 \cot \alpha \quad (2-277)$$

The coordinate of point B is as:

$$\frac{F_r}{P} = 1$$

$$\frac{F_a \cot \alpha}{P} = \xi$$

Putting into the equation (2-274) we can get:

$$X = 1 - Ye \quad (2-278)$$

For radial ball bearing $\xi = 1.1$

For angular contact ball bearing $\xi = 1.25$

For radial roller bearing $\xi = 1.5$

3. Radial and angular contact ball bearing dynamic equivalent load

There is no relation between contact angle and bearing load for most of bearing types. But for radial and angular contact ball bearings, their contact angle changes with bearing load. Especially while the nominal contact angle is very small or bearing is only subjecting to pure axial load, the contact angle change is more notable. Therefore while determining the dynamic equivalent load of these type bearings, we should consider the contact angle change.

When subjecting to radial load and axial load at the same time, for radial and angular contact ball bearing, the contact angle changes with rolling element location. In order to simplify the dynamic equivalent load P calculation, while determining these type ball bearing dynamic equivalent load, first we calculate the actual contact angle α under an axial load, then putting the value of the actual contact angle α into the equations (2-277) and (2-278) to get the X , Y , and e value.

The actual contact angle α calculation equation for radial and angular contact ball bearing is as:

For radial ball bearing:

$$\tan \alpha = \left(\frac{2k}{q} \right)^{\frac{3}{8}} \left(\frac{F_a}{zD_w^2} \right)^{\frac{1}{4}} \quad (2-155)$$

For angular contact ball bearing:

$$\left(\frac{\cos \alpha_0}{\cos \alpha} - 1 \right) \sin^{\frac{2}{3}} \alpha = \frac{k}{q} \left(\frac{F_a}{zD_w^2} \right)^{\frac{2}{3}} \quad (2-154)$$

$$q = f_i + f_e - 1 \quad (2-152)$$

Where α_0 — nominal contact angle (initial contact angle);

α — actual contact angle under axial load;

k — elastic deformation constant, checked out from Table 2-7

or Figure 2 - 16;

F_a — axial load (N);

D_w — rolling bearing diameter (mm);

Z — rolling element number.

In China, for the standard design of single row radial ball bearing, we usually take:

$$f_i = f_e = 0.515$$

But usually taking positive tolerance for groove curvature radius, actually the f is between 0.515 ~ 0.52. If taking the average value:

$$f_i = f_e = 0.5175$$

We can check out from Figure 2 - 16:

$$k = 0.000437$$

From the equation (2 - 152) we can get:

$$q = 0.035$$

In the equation (2 - 155), Z and D_w both are the interior structure dimensions that can not be found from general catalog and handbook. We can find the basic static load rating C_0 value in these book, but

$$C_0 = 1.25zD_w^2$$

Putting k , q , C_0 into the equation (2 - 155) we can get:

$$\tan\alpha \doteq 0.47 \left(\frac{F_a}{C_0} \right)^{\frac{1}{4}} \quad (2 - 279)$$

Putting the equation (2 - 279) into the equations (2 - 277) and (2 - 278), we can get the single row radial ball bearing e , X , Y values:

$$e = 1.1 \tan\alpha = 0.517 \left(\frac{F_a}{C_0} \right)^{\frac{1}{4}}$$

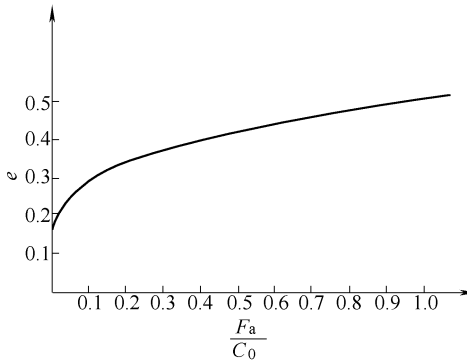
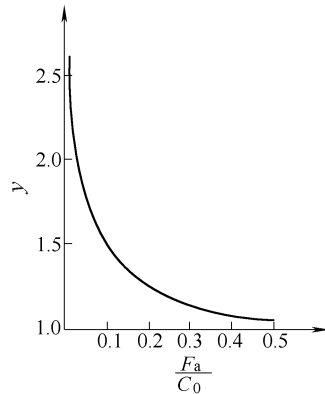
$$Y = 0.4 \cot\alpha = 0.851 \left(\frac{F_a}{C_0} \right)^{-\frac{1}{4}}$$

$$X = 0.56$$

In commonly used F_a/C_0 range, the Y and e values are listed in Table 2 - 26 and Figure 2 - 45, Figure 2 - 46.

Table 2-26 Single row radial ball bearing e , Y values

$\frac{F_a}{C_0}$	0.015	0.025	0.04	0.07	0.13	0.25	0.50
e	0.18	0.21	0.23	0.27	0.31	0.37	0.44
Y	2.43	2.14	1.90	1.66	1.42	1.20	1.01

**Figure 2-45 Single row radial ball bearing e value****Figure 2-46 Single row radial ball bearing Y value**

The X , Y , e calculation equations for every single row bearing are listed in Table 2-27.

Table 2-27 Single row bearing α , e , X , Y calculation equation

Bearing type	Single row radial ball bearing	Angular contact ball bearing ^①	Aligning ball bearing	Radial roller bearing
α	$\left(\frac{2k}{q}\right)^{\frac{3}{8}} \left(\frac{F_a}{zD_w^2}\right)^{\frac{1}{4}}$	$\left(\frac{\cos\alpha_0}{\cos\alpha} - 1\right) \sin^{\frac{2}{3}} \alpha$ $= \frac{k}{q} \left(\frac{F_a}{zD_w^2}\right)^{\frac{2}{3}}$	α	α
e	$1.1 \tan\alpha$	$1.25 \tan\alpha$	$1.5 \tan\alpha$	$1.5 \tan\alpha$
X	0.56	$\frac{3 - 2 \sin\alpha}{6 - 2 \sin\alpha}$	0.4	0.4
Y	$\frac{0.44}{e}$	$\frac{1 - X}{e}$	$0.4 \cot\alpha$	$0.4 \cot\alpha$

① When subjecting to axial load, for angular contact ball bearing, because of manufactur-

ing error (such as raceway wall thickness error, raceway axial runout), the actual dynamic equivalent load is higher than the theoretical calculation value. Usually introducing an experience coefficient η in axial factor Y , $\eta = 1 - 1/3 \sin \alpha$. So for angular contact ball bearing:

$$Y = \frac{0.4 \cot \alpha}{1 - \frac{1}{3} \sin \alpha}$$

4. Double row angular contact ball bearing dynamic equivalent load

Double row bearing discussing here designates the symmetrical, double row angular contact bearing with a contact angle α , such as aligning ball bearing, aligning roller bearing, double row tapered roller bearing, angular contact ball bearing in pair arrangement (except outer ring back face to front face that is tandem arrangement).

(1) Double row bearing “average” load

Same as the single row bearing, assume that the inner ring rotates relative to the load vector and the outer ring is stationary relative to the load vector. Lower corner symbol I, II express row I and row II respectively.

S_i expresses the entire probability of survival; S_{iI} and S_{iII} express the probability of survival for inner ring row I and row II respectively. From the equation (2-207) we can get:

$$\lg \frac{1}{S_{iI}} = k_{iI} Q_{iI}^w L^e \quad (2-280)$$

$$\lg \frac{1}{S_{iII}} = k_{iII} Q_{iII}^w L^e \quad (2-281)$$

For the entire inner ring:

$$S_I = S_{iI} S_{iII}$$

$$\lg \frac{1}{S_i} = \lg \frac{1}{S_{iI} S_{iII}} = k_i (Q_{iI}^w + Q_{iII}^w) L^e = k_i Q_i^w L^e \quad (2-282)$$

Where
$$Q_i = (Q_{iI}^w + Q_{iII}^w)^{\frac{1}{w}} \quad (2-283)$$

Using T_I , T_{II} expresses the load distribution factor in row I and row II respectively. From the equation (2-211) we can get:

$$Q_{iI} = Q_{\max I} J_1 (T_I)$$

$$Q_{iII} = Q_{\max II} J_1 (T_{II})$$

So

$$Q_i = Q_{\max I} \left\{ [J_1(T_I)]^w + \left(\frac{Q_{\max II}}{Q_{\max I}} \right)^w [J_1(T_{II})]^w \right\}^{\frac{1}{w}}$$

$$= Q_{\max I} J_1(T_I T_{II}) \quad (2-284)$$

$$J_1(T_I T_{II}) = \{ [J_1(T_I)]^w \left(\frac{Q_{\max II}}{Q_{\max I}} \right)^w [J_1(T_{II})]^w \}^{\frac{1}{w}} \quad (2-285)$$

For stationary outer ring, similarly we can get:

$$Q_e = Q_{\max I} J_2(T_I T_{II}) \quad (2-286)$$

$$J_2(T_I T_{II}) = \left\{ [J_2(T_I)]^w + \left(\frac{Q_{\max II}}{Q_{\max I}} \right)^w [J_2(T_{II})]^w \right\}^{\frac{1}{w}}$$

$$(2-287)$$

Q_i , Q_e are the “average” loads for double row bearing inner ring, outer ring. If putting the equation (2-102) into the equations (2-284) and (2-286), then we can get:

$$Q_i = \frac{F_r}{z \cos \alpha} \frac{J_1(T_I T_{II})}{J_r(T_I T_{II})} = \frac{F_a}{z \sin \alpha} \frac{J_1(T_I T_{II})}{J_a(T_I T_{II})} \quad (2-288)$$

$$Q_e = \frac{F_r}{z \cos \alpha} \frac{J_2(T_I T_{II})}{J_r(T_I T_{II})} = \frac{F_a}{z \sin \alpha} \frac{J_2(T_I T_{II})}{J_a(T_I T_{II})} \quad (2-289)$$

$J_1(T_I T_{II})$ and $J_2(T_I T_{II})$ values are listed in Table 2-28.

Table 2-28 Double row bearing $J_1(T_I T_{II})$, $J_2(T_I T_{II})$

Load distribution factor		Point contact		Line contact	
T_I	T_{II}	J_1	J_2	J_1	J_2
0.5	0.5	0.6925	0.7233	0.7577	0.7867
0.6	0.4	0.5983	0.6231	0.6807	0.7044
0.7	0.3	0.5986	0.6215	0.6806	0.7032
0.8	0.2	0.6105	0.6331	0.6907	0.7127
0.9	0.1	0.6248	0.6453	0.7028	0.7229
1.0	0	0.6372	0.6566	0.7132	0.7323
1.25	0	0.6652	0.6821	0.7366	0.7532
1.67	0	0.7064	0.7190	0.7705	0.7832
2.5	0	0.7707	0.7777	0.8216	0.8301
5	0	0.8675	0.8693	0.8989	0.9014
∞	0	1	1	1	1

(2) Double row bearing dynamic equivalent load

Same as single row bearing, also we can make the relationship of F_r/P_1 and $F_a \cot \alpha / P_1$ for double row bearing, as shown in Figure 2-47 and Figure 2-48.

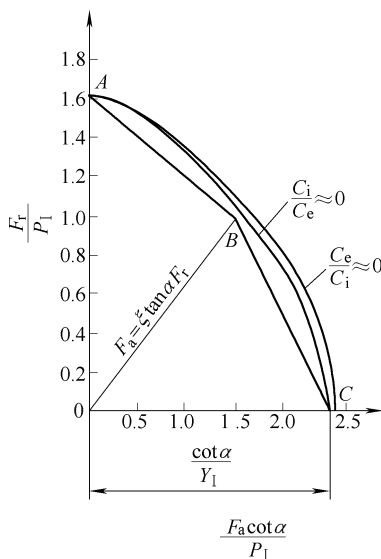


Figure 2-47 F_r and F_a
relationship for point contact.

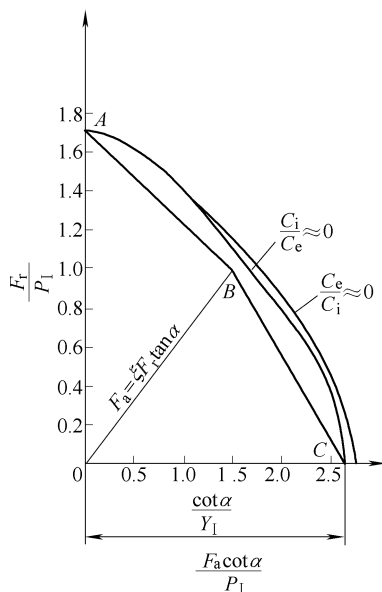


Figure 2-48 F_r and F_a
relationship for line contact.

In the figure:

F_r — radial load on double row bearing;

F_a — axial load on double row bearing;

P_1 — dynamic equivalent load for single row bearing;

Y_1 — single row bearing axial factor.

The curves in Figure 2-47 and Figure 2-48 also express the relationship of radial load F_r and axial load F_a for getting the same dynamic equivalent load P .

In order to simplify the dynamic equivalent load P calculation, in these Figures we use straight lines AB and BC instead of curve line. The equation for line AB and BC can write as a uniform equation:

$$P = XF_r + YF_a \quad (2-274)$$

But we will take different X , Y value for lines AB part and BC part.

1) X , Y value for line BC

The coordinates of the intersection point B of line AB and line BC are:

$$\left. \begin{aligned} \frac{F_r}{P_1} &= 1 \\ \frac{F_a \cot \alpha}{P_1} &= 1.5 \end{aligned} \right\}$$

So, we can get the relationship of F_r and F_a at point B as:

$$\frac{F_a}{F_r} = e = 1.5 \tan \alpha$$

When $F_a/F_r > e = 1.5 \tan \alpha$, we should calculate the dynamic equivalent load P by using the equation of line BC part.

Double row bearing which contact angle is α can carry any direction load. With the change of the applied load direction, the load distribution in the two rows also changes. If a pure radial load is applied to the bearing, the two rows are subjecting to the same load. If radial load and axial load are applied to the bearing at the same time, one row carrying load area increases and the other row carrying load area decreases with loading angle increasing. From Table 2-13 we can know when

For ball bearing $F_a > 1.67 F_r \tan \alpha$

For roller bearing $F_a > 1.91 F_r \tan \alpha$

there is only one row rolling element carrying load, and the other row rolling element is not carrying any load. At this time, this bearing life can be calculated from single row bearing, or from double row bearing. The result should be the same.

If P_1 , C_1 , X_1 , Y_1 express the values of single row bearing respectively, and P , C , X , Y express the values of double row bearing respectively, then we can get:

$$L = \left(\frac{C_1}{P_1} \right)^{\epsilon} = \left(\frac{C}{P} \right)^{\epsilon}$$

$$\text{So} \quad \frac{P}{P_1} = \frac{C}{C_1} = 2^{1 - \frac{1}{\epsilon}} \quad (2-290)$$

For ball bearing: $P = 1.62 P_1$

For roller bearing: $P = 1.71 P_1$

But
$$P_1 = X_1 F_r + Y_1 F_a$$
$$P = XF_r + YF_a$$

Putting the equation (2-290) into the above equations we can get:

$$X = 2^{1-\frac{1}{w}} X_1 \quad (2-291)$$

$$Y = 2^{1-\frac{1}{w}} Y_1 \quad (2-292)$$

Putting the equations (2-277) and (2-278) into the above equations we can get:

For ball bearing:

$$\left. \begin{aligned} X &= 1.62 * (1 - 0.4 \cot \alpha * 1.5 \tan \alpha) = 0.65 \\ Y &= 0.65 \cot \alpha \end{aligned} \right\} (2-293)$$

For roller bearing:

$$\left. \begin{aligned} X &= 0.67 \\ Y &= 0.67 \cot \alpha \end{aligned} \right\} (2-294)$$

2) X, Y value for line AB

When $F_a/F_r \leq 1.5 \tan \alpha$, we should calculate the dynamic equivalent load P by using the equation of line AB . For determining X, Y values, we should use points A and B coordinates.

For point A , we can get;

$$\frac{F_r}{P_1} = 2^{1-\frac{1}{w}} \frac{F_a \cot \alpha}{P_1} = 0$$

So
$$F_r = 2^{1-\frac{1}{w}} P_1$$

From the equation (2-290) we can get:

$$P = F_r$$

Therefore we can get: $X = 1$

For point B , we can get:

$$\frac{F_r}{P_1} = 1$$

$$F_a = 1.5 \tan \alpha F_r$$

Putting the above values into the equation (2-274) we can get:

$$2^{1-\frac{1}{w}} = X + 1.5 \tan \alpha Y$$

So

$$Y = \frac{2^{1-\frac{1}{w}} - 1}{1.5} \cot \alpha \quad (2-295)$$

For ball bearing we can get:

$$\left. \begin{aligned} X &= 1 \\ Y &= 0.42 \cot \alpha \end{aligned} \right\} \quad (2-296)$$

For roller bearing we can get:

$$\left. \begin{aligned} X &= 1 \\ Y &= 0.45 \cot \alpha \end{aligned} \right\} \quad (2-297)$$

Summarizing the above mentioned, the equations for X , Y , e values of double row angular contact ball bearing are listed in Table 2-29.

Table 2-29 Double row angular contact ball bearing X , Y , e values

Bearing type	$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		e
	X	Y	X	Y	
aligning ball bearing	1	$0.42 \cot \alpha$	0.65	$0.65 \cot \alpha$	$1.5 \tan \alpha$
aligning roller bearing	1	$0.45 \cot \alpha$	0.67	$0.67 \cot \alpha$	$1.5 \tan \alpha$
double row tapered roller bearing	1	$0.45 \cot \alpha$	0.67	$0.67 \cot \alpha$	$1.5 \tan \alpha$

While determining the dynamic equivalent load P for double row angular contact ball bearing, we should consider the contact angle changes. First we calculate the single row bearing X_1 , Y_1 values, then we calculate the double row bearing X , Y values by using the equations (2-291) and (2-92).

5. Thrust bearing dynamic equivalent load

Thrust and thrust angular contact bearing basic dynamic load rating designates the central axial load. For the thrust bearings carrying axial load and radial load at the same time, first we change the actual load to a dynamic equivalent load P , and then the bearing life can be calculated.

For thrust bearing of $\alpha = 90^\circ$, it can not carry radial load. So the dynamic equivalent load of this type bearing is:

$$P = F_a \quad (2-298)$$

But thrust radial bearing, such as thrust aligning roller bearing, can carry certain amount of radial load. But this type bearing mainly is to carry axial load, and its basic dynamic load rating designates central axial load. This type bearing can also be considered as radial angular contact bearing which contact angle α is larger, which radial dynamic equivalent load can be calculated by the equation (2-274). If P_r expresses this type bearing radial dynamic equivalent load, and its radial factor is expressed by X_r , from the equation (2-274) we can get:

$$P_r = X_r F_r + Y F_a \quad (2-299)$$

The bearing life should be the same calculated by using radial dynamic equivalent load P_r and by axial dynamic equivalent P , that is:

$$L = \left(\frac{C_r}{P_r} \right)^{\epsilon} = \left(\frac{C}{P} \right)^{\epsilon}$$

So
$$\frac{C_r}{P_r} = \frac{C}{P}$$

For a bearing, the relationship between the load rating expressed by radial load and by axial load should be:

$$C_r = Y C_a$$

Therefore, the dynamic equivalent load expressed by central axial load is:

$$P = \frac{P_r}{Y} = \frac{X_r}{Y} F_r + F_a \quad (2-300)$$

From the equations (2-277) and (2-278) we can get:

$$P = X F_r + F_a \quad (2-301)$$

$$X = \tan \alpha \quad (2-302)$$

6. The influence for the dynamic equivalent load when outer ring rotating relative to load vector

It is assumed in the above mentioned for determining the basic dynamic load rating C and dynamic equivalent load P : that the inner ring is rotating and the outer ring is in stationary relative to the load vector. Under this assuming condition, the maximum rolling element load is always applied on the same point of outer ring raceway and lighter load or completely no load is applied to the other points of the outer ring raceway. But every point on the inner ring raceway carries the same altering load. While one point of inner

ring raceway enters the loading area, it firstly will begin contacting with very lighter loaded rolling element, then it will continue contacting with heavier loaded rolling element until contacting with the maximum loaded rolling element, and after that this point will gradually carry lighter rolling element load.

If the bearing condition is opposite with the above assuming condition, that is the outer ring is rotating and the inner ring is in stationary relative to the load vector, the calculation method of average rolling element load will be different from the above mentioned.

If under this assuming condition, Q_i , Q_e expresses the inner ring, outer ring average rolling element load respectively; C_i , C_e expresses the inner ring, outer ring basic load rating. From the equations (2-205) and (2-209) we can get:

$$Q'_i = Q_e = Q_{\max} J_2 \quad (2-303)$$

$$Q'_e = Q_i = Q_{\max} J_1 \quad (2-304)$$

Similarly from the equations (2-213) and (2-214) we can get:

$$C'_i = C_e = Q_{ce} z \cos \alpha \frac{J_r(0.5)}{J_2(0.5)} \quad (2-305)$$

$$C'_e = C_i = Q_{ci} z \cos \alpha \frac{J_r(0.5)}{J_1(0.5)} \quad (2-306)$$

In two situations that the inner ring is rotating relative to load vector and in stationary relative to load vector, the inner ring raceway load rating should be the same value, so:

$$\frac{C'_i}{C_i} = \frac{J_1(0.5)}{J_2(0.5)} \quad (2-307)$$

Similarly we can get:

$$\frac{C'_e}{C_e} = \frac{J_2(0.5)}{J_1(0.5)} \quad (2-308)$$

If let:

$$v = \frac{J_2(0.5)}{J_1(0.5)} \quad (2-309)$$

Then we can get:

$$C'_i = \frac{1}{v} C_i \quad (2-310)$$

$$C_e' = vC_e \quad (2-311)$$

In load distribution factor $T = 0.5$, putting the equations (2-310) and (2-311) into the equation (2-272) we can get:

$$\frac{F_r}{P} = \left[\left(\frac{vC}{C_i} \right)^w + \left(\frac{C}{vC_e} \right)^w \right]^{-\frac{1}{w}} \quad (2-312)$$

If let:

$$V = \left[\left(\frac{vC}{C_i} \right)^w + \left(\frac{C}{vC_e} \right)^w \right]^{-\frac{1}{w}} \quad (2-313)$$

We can get:

$$P = V F_r \quad (2-314)$$

Where V is the rotating factor.

If putting the equations (2-224) and (2-225) into the equation (2-313) we can get:

$$V = v \left[\frac{1 + \left(\frac{C_i}{v^2 C_e} \right)^w}{1 + \left(\frac{C_i}{C_e} \right)^w} \right]^{\frac{1}{w}} \quad (2-315)$$

The V values of different C_i/C_e values are listed in Table 2-30.

Table 2-30 Rotating factor V

$\frac{C_i}{C_e}$		0	0.5	0.75	1	1.5	2	∞
V	point contact	1.044	1.037	1.022	1.003	0.976	0.966	0.958
	line contact	1.038	1.035	1.024	1.003	0.975	0.967	0.963

From the table values we can know that the maximum rotating factor V is 1.044 (point contact) and 1.038 (line contact). But in actual bearing, the strength of inner ring and outer ring can not reach such values. So in actual calculation if taking $V = 1$, then V can be omitted in the dynamic equivalent load equation (2-274).

2.7 International Standard ISO 281 and China National Standard GB/T6391

The Lundberg-Palmgren rolling bearing life theory introduced in this chapter 2.6, is applied to bearing load capacity and bearing life calculations under varied design parameters, the base of the International Standard ISO 281. Through over ten years of investigation, ISO/TC4 (International standard organization fourth committee) published a recommended standard ISO/R281 in 1962, and through many times of revision for another twenty years, published the official standard ISO 281: 1990. China accepted it equally as National Standard GB/T 6391-2003. This standard is applicable to the rolling bearings which dimensions are in the range of national standard stipulation, made by high quality hardening rolling bearing steel commonly used not contemporary era and good manufacturing methods, and which contact surface shapes conform with common rolling bearing design.

The terms and definitions using in this standard are as follows:

1. Life

(of an individual rolling bearing) Numbers of rotation of one of bearing rings or washers in relation to the other ring or washer before the first evidence of fatigue develops in the material of one of the rings or washers or one of the rolling elements.

Note: Life may also be expressed in number of hours of operation at a given constant speed of rotation.

2. Reliability

(in the context of bearing life) For a group of apparently identical rolling bearing, operating under the same conditions, the percentage of the group that is expected to attain or exceed a specified life.

Note: The reliability of an individual rolling bearing is the probability that the bearing will attain or exceed a specified life.

3. Rating life

Predicted value of life based on a basic dynamic radial load rating or a basic dynamic axial load rating

4. Basic rating life

Rating life associated with 90% reliability for bearing manufactured with commonly used high quality material of good manufacturing quality, and operating under conventional operating conditions.

5. Modified rating life

Rating life modified for 90% or other reliability, bearing fatigue load, and/or contaminated lubricant, and/or other non- conventional operation conditions.

Note: The term “modified rating life” is new in this document and replaces “adjusted rating life”.

6. Basic dynamic radial load rating

Constant stationary radial load which a rolling bearing can theoretically endure for a basic rating life of one million revolutions.

Note: In the case of a single-row angular contact bearing, the radial load rating refers to the radial component of that load which causes a purely radial displacement of the bearing rings in relation to each other.

7. Basic dynamic axial load rating

Constant central axial load which a rolling bearing can theoretically endure for a basic rating life of one million revolutions.

8. Dynamic equivalent radial load

Constant central radial load under the influence of which a rolling bearing would have the same life as it would attain under the actual load conditions.

9. Dynamic equivalent axial load

Constant central axial load under the influence of which a rolling bearing would have the same life as it would attain under the actual load conditions.

10. Roller diameter

(applicable in the calculation of load rating) Theoretical diameter in a radial plane through the middle of the roller length for a symmetrical roller.

Note 1: For a tapered roller, the applicable diameter is equal to the mean value of the diameter at the imaginary sharp corners at the large end and at the small end of the roller.

Note 2: For an asymmetrical convex roller, the applicable diameter is an approximation of

the diameter at the point of contact between the roller and the ribless raceway at zero load.

11. Effective roller length

(applicable in calculation of load rating) Theoretical maximum length of contact between a roller and that raceway where the contact is shortest.

Note: This is normally taken to be either the distance between the theoretical sharp corner of the roller minus the roller chamfers or the raceway width, excluding the grinding undercuts, whichever is the smaller.

12. Nominal contact angle

Angle between a plane perpendicular to a bearing axis (a radial plane) and the normal line of action of the resultant of the forces transmitted by a bearing ring or washer to a rolling element.

Note: For bearings with asymmetrical rollers, the nominal contact angle is determined by the contact with the non-ribbed raceway.

13. Pitch diameter of ball set

Diameter of the circle containing the centers of the balls in one row in a bearing.

14. Pitch diameter of roller set

Diameter of the circle intersecting the roller axes at the middle of the rollers in one row in a bearing.

15. Conventional operating conditions

Conditions which may be assumed to prevail for a bearing which is properly mounted and protected from foreign matter, adequately lubricated, conventionally loaded, not exposed to extreme temperature and not run at exceptionally low or high speed

Symbols

C_r — basic dynamic radial load rating (N);

C_a — basic dynamic axial load rating (N);

C_{or} — basic static radial load rating^① (N);

C_{oa} — basic static axial load rating^① (N);

Note: ① Its definition and calculation method and values are given in GB/T 4662 — 2006 and ISO76; 2006.

D_w — ball bearing diameter (mm);

D_{we} — roller diameter applicable in the calculation of load rating (mm);

D_{pw} — pitch diameter of ball or roller set (N);

F_r — bearing radial load = radial component of actual bearing load (N);

F_a — bearing axial load = axial component of actual bearing load (N);

L_{10} — basic rating life (million revolutions);

L_{na} — modified basic rating life (million revolutions) (definition of 1995 version);

L_{nm} — modified basic rating life (million revolutions);

L_{we} — roller length applicable in the calculation of load rating (mm)

P_r — dynamic equivalent radial load (N);

P_a — dynamic equivalent axial load (N);

X — dynamic radial load factor;

Y — dynamic axial load factor;

z — number of rolling elements in a single-row bearing; number of rolling elements per row of a multiple row bearing with the same number of rolling elements per row;

b_m — rating factor for contemporary, commonly used, high quality hardened bearing steel in accordance with good manufacturing practices of which varies with bearing type and design;

e — limiting value of F_a/F_r for the applicability of different values of factors X and Y ; Weibull exponent;

f_c — factor which depends on the geometry of the bearing components, the accuracy to which the various components are made, and the material;

f_0 — factor which depends on the geometry of the bearing components, the stress level;

i — number of rows of rolling elements;

α — nominal contact angle ($^\circ$);

- a_{iso} — life modification factor for reliability of determining of the stipulating system method in Section 2.7.5;
- a_1 — life modification factor for reliability;
- n — subscript for probability of failure (%);
- S — reliability (probability of survival) (%);
- k — viscosity ratio of actual kinematic oil viscosity at operating temperature divided by the reference kinematic viscosity for adequate lubrication;
- λ — film parameter of ratio of lubricant film thickness to composite r. m. s surface roughness, used to estimate the influence of lubrication on bearing life;
- v — actual kinematic viscosity at the operating temperature, in square millimeters;
- v_1 — reference kinematic viscosity required to obtain adequate lubrication, in square millimeters;
- σ — (real) stress, used in fatigue criterion (MPa);
- σ_u — fatigue stress limit of raceway material, used in fatigue criterion (MPa).

2.7.1 Radial Ball Bearings

1. Basic dynamic radial load rating

The basic dynamic radial load rating C_r for radial contact and angular contact ball bearing is given by the equations:

$$\text{When } D_w \leq 25.4 \text{ mm, } C_r = b_m f_c (\cos \alpha)^{0.7} Z^{2/3} D_w^{1.8}$$

$$\text{When } D_w > 25.4 \text{ mm, } C_r = 3.647 b_m f_c (\cos \alpha)^{0.7} Z^{2/3} D_w^{1.4}$$

The values of b_m and f_c are listed in Table 2 – 31 and 2 – 32 respectively, which apply to the radial and angular contact ball bearings with a cross-sectional raceway groove curvature radius of the inner ring not larger than $0.52D_w$ and that of the outer ring not larger than $0.53D_w$ and to the self-aligning ball bearings with a raceway groove curvature radius of the inner ring not larger than $0.53D_w$.

The load — carrying capability of a bearing is not necessarily increased

by the use of a smaller groove curvatures radius. But it is reduced by the use of a groove curvature radius larger than those above indicated.

Table 2 – 31 Values of b_m for radial ball bearing

Bearing type	b_m
radial and angular contact ball bearings (except filling slot bearings) , self-aligning ball bearings	1. 3
filling slot bearings	1. 1
insert bearing	1. 3

Table 2 – 32 Values of factor f_c for radial ball bearings

$\frac{D_w \cos \alpha^{①}}{D_{pw}}$	single-row radial contact ball bearings and single-row and double-row angular contact ball bearings	double-row radial contact ball bearing	single-row and double-row self-aligning ball bearings	single-row radial contact separable ball bearings (magneto bearings)
0. 01	29. 1	27. 5	9. 9	9. 4
0. 02	35. 8	33. 9	12. 4	11. 7
0. 03	40. 3	38. 2	14. 3	13. 4
0. 04	43. 8	41. 5	15. 9	14. 9
0. 05	46. 7	44. 2	17. 3	16. 2
0. 06	49. 1	46. 5	18. 6	17. 4
0. 07	51. 1	48. 4	19. 9	18. 6
0. 08	52. 8	50	21. 1	19. 5
0. 09	54. 3	51. 4	22. 3	20. 6
0. 1	55. 5	52. 6	23. 4	21. 5
0. 11	56. 6	53. 6	24. 5	22. 5
0. 12	57. 5	54. 5	25. 6	23. 4
0. 13	58. 2	55. 2	26. 6	24. 4
0. 14	58. 8	55. 7	27. 7	25. 3
0. 15	59. 3	56. 1	28. 7	26. 2
0. 16	59. 6	56. 5	29. 7	27. 1
0. 17	59. 8	56. 7	30. 7	27. 9
0. 18	59. 9	56. 8	31. 7	28. 8
0. 19	60	56. 8	32. 6	29. 7

(continued)

$\frac{D_w \cos \alpha^{\text{①}}}{D_{pw}}$	single-row radial contact ball bearings and single-row and double-row angular contact ball bearings	double-row radial contact ball bearing	single-row and double-row self-aligning ball bearings	single-row radial contact separable ball bearings (magneto bearings)
0.2	59.9	56.8	33.5	30.5
0.21	59.8	56.6	34.4	31.3
0.22	59.6	56.5	35.2	32.1
0.23	59.3	56.2	36.1	32.9
0.24	59	55.9	36.8	33.7
0.25	58.6	55.5	37.5	34.5
0.26	58.2	55.1	38.2	35.2
0.27	57.7	54.6	38.8	35.9
0.28	57.1	54.1	39.4	36.6
0.29	56.6	53.6	39.9	37.2
0.3	56	53	40.3	37.8
0.31	55.3	52.4	40.6	38.4
0.32	54.6	51.8	40.9	38.9
0.33	53.9	51.1	41.1	39.4
0.34	53.2	50.4	41.2	39.8
0.35	52.4	49.7	41.3	40.1
0.36	51.7	48.9	41.3	40.4
0.37	50.9	48.2	41.2	40.7
0.38	50	47.4	41	40.8
0.39	49.2	46.6	40.7	40.9
0.4	48.4	45.8	40.4	40.9

Note: ① Values of f_c for intermediate values of $\frac{D_w \cos \alpha}{D_{pw}}$ are obtained by linear interpolation.

2. Basic dynamic radial load rating for bearing combinations

When calculating the basic dynamic radial load rating for two identical single-row radial contact (groove type) ball bearings mounted side by side on the same shaft, operated as a unit (paired mounting), the pair is considered as one double-row radial contact ball bearing.

When calculating the basic dynamic radial load rating for two identical single-row angular contact ball bearings mounted side by side on the same shaft, operated as a unit (paired mounting) in a back-to-back or a face-to-

face arrangement, the pair is considered as one double-row angular contact ball bearing.

The basic dynamic radial load rating, for two or more identical radial contact ball bearings or two or more identical angular contact ball bearings mounted side by side on the same shaft, operated as a unit (paired or stack mounting) in a tandem arrangement, is the number of bearing pairs to the power of 0.7 times the basic dynamic radial load rating of one single-row bearing.

The proper manufacturing and mounting can ensure even load distribution between the bearings.

If, for some technical reasons, the bearing arrangement is regarded as a number of pairs of single-row specially manufactured bearings which are replaceable independently of each other, then the above mentioned stipulation does not apply.

3. Dynamic equivalent radial load

The dynamic equivalent radial load for radial and angular contact ball bearings, under constant radial and axial loads, is given by

$$P_r = XF_r + YF_a$$

Where the values of factors X and Y are given in Table 2-33.

4. Dynamic equivalent radial load for bearing combinations

When calculating the equivalent radial load for two identical single-row angular contact ball bearings mounted side by side on the same shaft, operated as a unit (paired mounting) in a back-to-back or a face-to-face arrangement, the pair is considered as one double-row angular contact ball bearing.

When calculating the equivalent radial load for two or more identical radial contact ball bearings or two or more identical angular contact ball bearings mounted side by side on the same shaft, operated as a unit (paired or stack mounting) in a tandem arrangement, the values of X and Y for a single-row bearing should be used.

The "relative axial load" (see Table 2-33) is established by using $i = 1$ and the F_a and C_{0r} values which both refer to one pair of the bearings only (even though the F_r and F_a values referring to the total loads are used for the calculation of the equivalent load for the entire arrangement).

Table 2 – 33 Values of X and Y for radial ball bearings

Bearing type	“Relative axial load ” ①②		Single-row bearing				Double-row bearing				<i>e</i>																																																																																														
			$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$																																																																																																
			<i>X</i>	<i>Y</i>	<i>X</i>	<i>Y</i>	<i>X</i>	<i>Y</i>	<i>X</i>	<i>Y</i>																																																																																															
radial contact ball bearings	$\frac{f_0 F_a^{(3)}}{C_{0r}}$	$\frac{F_a}{iZD_w^2}$	1	0	0.56	1.45	1	0	0.56	1.45	0.3																																																																																														
	0.172	0.172										2.3	1.99	1.71	1.55	1.31	1.15	1.04	1																																																																																						
	0.345	0.345																																																																																																							
	0.689	0.689																																																																																																							
	1.03	1.03																																																																																																							
	1.38	1.38																																																																																																							
	2.07	2.07																																																																																																							
	3.45	3.45																																																																																																							
	5.17	5.17																																																																																																							
6.89	6.89																																																																																																								
angular contact ball bearings	$\frac{f_0 i F_a^{(3)}}{C_{0r}}$	$\frac{F_a}{ZD_w^2}$	1	0	For this type, use the <i>X</i> , <i>Y</i> and <i>e</i> values applic- able to single- row radial co- ntact ball be- arings.	1	1.75	0.78	2.36	1.87	1.69																																																																																														
	0.173	0.172										2.78	3.74	2.07	2.52	2.13	1.87	1.69	1.63																																																																																						
	0.346	0.345																		2.4	3.23	1.87	2.13	1.87	1.69	1.63																																																																															
	0.692	0.689																									2.07	2.78	1.87	2.13	1.87	1.69	1.63																																																																								
	1.04	1.03																																1.87	2.52	1.87	1.69	1.63																																																																			
	1.38	1.38																																					1.75	2.36	1.58	2.13	1.87	1.69	1.63																																																												
	2.08	2.07																																												1.58	2.13	1.87	1.69	1.63																																																							
	3.46	3.45																																																	1.39	1.87	1.69	1.63																																																			
	5.19	5.17																																																					1.26	1.69	1.63																																																
	6.92	6.89																																																								1.21	1.63	0.52																																													
	$\alpha = 5^\circ$	0.175																																																											0.172	1	0	0.46	1.88	1	1.55	0.75	2.18	3.06	0.29																																		
		0.35																																																											0.345											1.71	1.98	1.76	1.63	2.18	0.4	0.44	0.49	0.54																									
		0.7																																																											0.689																				1.52	1.76	1.63	2.29	0.38																				
		1.05																																																											1.03																									1.41	1.63	2.29	0.38																
		1.4																																																											1.38																													1.34	1.23	1.1	1.01	1											
		2.1																																																											2.07																																		1.23	1.42	1.27	1.79	0.49	0.54					
		3.50																																																											3.45																																								1.1	1.01	1		
		5.25																																																											5.17																																											1.01	1
		7																																																											6.89																																												
	$\alpha = 10^\circ$	0.175																																																											0.172	1	0	0.46	1.88	1	1.55	0.75	2.18	3.06	0.29																																		
		0.35																																																											0.345											1.71	1.98	1.76	1.63	2.18	0.4	0.44	0.49	0.54																									
0.7		0.689	1.52	1.76	1.63	2.29	0.38																																																																																																		
1.05		1.03						1.41	1.63	2.29	0.38																																																																																														
1.4		1.38										1.34	1.23	1.1	1.01	1																																																																																									
2.1		2.07															1.23	1.42	1.27	1.79	0.49	0.54																																																																																			
3.50		3.45																					1.1	1.01	1																																																																																
5.25		5.17																								1.01	1																																																																														
7		6.89																										1	1																																																																												

(continued)

Bearing type		“Relative axial load ” ^{①②}		Single-row bearing				Double-row bearing				<i>e</i>
				$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		
				<i>X</i>	<i>Y</i>	<i>X</i>	<i>Y</i>	<i>X</i>	<i>Y</i>	<i>X</i>	<i>Y</i>	
angular contact ball bearing	$\alpha = 15^\circ$	0. 178	0. 172	1	0	0. 44	1. 47	1	1. 65	0. 72	2. 39	0. 38
		0. 357	0. 345				1. 4		1. 57		2. 28	0. 4
		0. 714	0. 689				1. 3		1. 46		2. 11	0. 43
		1. 07	1. 03				1. 23		1. 38		2	0. 46
		1. 43	1. 38				1. 19		1. 34		1. 93	0. 47
		2. 14	2. 07				1. 12		1. 26		1. 82	0. 5
		3. 57	3. 45				1. 02		1. 14		1. 66	0. 55
		5. 35	5. 17				1		1. 12		1. 63	0. 56
		7. 14	6. 89				1		1. 12		1. 63	0. 56
	$\alpha = 20^\circ$	—	—	1	0	0. 43	1	1. 09	0. 7	1. 63	0. 57	
	$\alpha = 25^\circ$	—	—			0. 41	0. 87	0. 92	0. 67	1. 41	0. 68	
	$\alpha = 30^\circ$	—	—			0. 39	0. 76	0. 78	0. 63	1. 24	0. 8	
	$\alpha = 35^\circ$	—	—			0. 37	0. 66	0. 66	0. 6	1. 07	0. 95	
	$\alpha = 40^\circ$	—	—			0. 35	0. 57	0. 55	0. 57	0. 93	1. 14	
	$\alpha = 45^\circ$	—	—			0. 33	0. 5	0. 47	0. 54	0. 81	1. 34	
self-aligning ball bearing				1	0	0. 4	0. 4 cot α	1	0. 42 cot α	0. 65	0. 65 cot α	1. 5 tan α
single-row radial contact separable ball bearings (magneto bearings)				1	0	0. 5	2. 5	—	—	—	—	0. 2

- Note: ① Permissible maximum value depends on the bearing design (internal clearance and raceway groove depth). Use the first or second column depending on available information.
- ② Values of *X*, *Y* and *e* for intermediate “ relative axial loads ” and/or contact angle are obtained by linear interpolation.
- ③ Values of *f*₀ see ISO 76 or GB/T4662—2003. Section 2.9 of this chapter will give introduction.

Editor notes: In Table 2 - 33 , the value of *e* in $\frac{f_0 i F_a^e}{C_{0r}}$ should be the Weibull exponent , for ball bearing $e = 10/9 = 1.11$, but $C_{0r} = f_0 i z D_w^2 \cos \alpha$ therefore , using F_a^e instead of $\cos \alpha$, to let the value of $\frac{f_0 i F_a^e}{C_{0r}}$ close to the value of $\frac{F_a}{z D_w^2}$.

5. Basic rating life

The basic rating life for a radial ball bearing is given by life equation:

$$L_{10} = \left(\frac{C_r}{P_r} \right)^3$$

The value of C_r and P_r are calculated in accordance with Section 2.7.1. The life equation is also used for the evaluation of the life of two or more single-row bearings operated as a unit, as referred in Section 2.7.1. In this case, the load rating C_r is calculated for the entire bearing arrangement and the equivalent load P_r is calculated for the total load acting on the arrangement, using the values of X and Y indicated in Section 2.7.1.

The life equation gives satisfactory results for a broad range of bearing loads. However, extra-heavy loads may cause detrimental plastic deformations at the ball/raceway contacts. The user should therefore consult the bearing manufacturer to establish the applicability of the life equation when P_r exceeds C_{or} or $0.5C_r$.

2.7.2 Thrust Ball Bearings

1. Basic dynamic axial load rating

(1) Single-row bearing

The basic dynamic axial load rating for single-row, single-direction or double-direction thrust ball bearings is given by:

When $D_w \leq 25.4\text{mm}$ and $\alpha = 90^\circ$, $C_a = b_m f_c Z^{2/3} D_w^{1.8}$

When $D_w \leq 25.4\text{mm}$ and $\alpha \neq 90^\circ$, $C_a = b_m f_c (\cos \alpha)^{0.7} \tan \alpha Z^{2/3} D_w^{1.8}$

When $D_w > 25.4\text{mm}$ and $\alpha = 90^\circ$, $C_a = 3.647 b_m f_c Z^{2/3} D_w^{1.4}$

When $D_w > 25.4\text{mm}$ and $\alpha \neq 90^\circ$, $C_a = 3.647 b_m f_c (\cos \alpha)^{0.7} \tan \alpha Z^{2/3} D_w^{1.4}$

where Z is the number of balls carrying the same direction load and $b_m = 1.3$.

Values of f_c are given in Table 2-34, applicable to the bearings with cross-sectional raceway groove curvature radii not larger than $0.54D_w$.

The load-carrying ability of a bearing is not necessarily increased by the use of a smaller groove curvature radius, but is reduced by the use of a

groove curvature radius larger than that indicated above. In the latter case, a correspondingly reduced value of f_c shall be used.

(2) Double-row or more-row bearings

The basic dynamic axial load rating for double-row or more-row thrust ball bearing carrying load in the same direction is given by:

$$C_a = (Z_1 + Z_2 + \dots + Z_n) \\ * \left[\left(\frac{Z_1}{C_{a1}} \right)^{10/3} + \left(\frac{Z_2}{C_{a2}} \right)^{10/3} + \dots + \left(\frac{Z_n}{C_{an}} \right)^{10/3} \right]^{-3/10}$$

If the ball numbers $Z_1, Z_2, Z_3, \dots, Z_n$ express each row rolling element numbers, the load rating $C_{a1}, C_{a2}, \dots, C_{an}$ can be calculated from the appropriate single-row bearing equation given in (1).

2. Dynamic equivalent axial load

The dynamic equivalent axial load for thrust ball bearing with $\alpha \neq 90^\circ$, under constant radial and axial loads, is given by:

$$P_a = XF_r + YF_a$$

Where the values of X and Y are given in Table 2-35.

Thrust ball bearing with $\alpha = 90^\circ$ can support axial load only. The dynamic equivalent axial load for this type of bearing is given by:

$$P_a = F_a$$

3. Basic rating life

(1) The basic rating life for a thrust ball bearing is given by the life equation:

$$L_{10} = \left(\frac{C_a}{P_a} \right)^3$$

C_a and P_a are calculated in accordance with the introduction method of the above mentioned.

(2) The life equation gives satisfactory result for a broad range of bearing loads. However, extra-heavy loads may cause detrimental plastic deformation at ball/raceway contacts. The user should therefore consult the bearing manufacturer to establish the applicability of the life equation when P_a exceeds $0.5C_a$.

Table 2 – 34 Thrust ball bearing f_c values

$\frac{D_w^{①}}{D_{pw}}$	f_c	$\frac{D_w \cos \alpha^{①}}{D_{pw}}$	f_c		
	$\alpha = 90^\circ$		$\alpha = 45^\circ ②$	$\alpha = 60^\circ$	$\alpha = 75^\circ$
0. 01	36. 7	0. 01	42. 1	39. 2	37. 3
0. 02	45. 2	0. 02	51. 7	48. 1	45. 9
0. 03	51. 1	0. 03	58. 2	54. 2	51. 7
0. 04	55. 7	0. 04	63. 3	58. 9	56. 1
0. 05	59. 5	0. 05	67. 3	62. 6	59. 7
0. 06	62. 9	0. 06	70. 7	65. 8	62. 7
0. 07	65. 8	0. 07	73. 5	68. 4	65. 2
0. 08	68. 5	0. 08	75. 9	70. 7	67. 3
0. 09	71	0. 09	78	72. 6	69. 2
0. 1	73. 3	0. 1	79. 7	74. 2	70. 7
0. 11	75. 4	0. 11	81. 1	75. 5	
0. 12	77. 4	0. 12	82. 3	76. 6	
0. 13	79. 3	0. 13	83. 3	77. 5	
0. 14	81. 1	0. 14	84. 1	78. 3	
0. 15	82. 7	0. 15	84. 7	78. 8	
0. 16	84. 4	0. 16	85. 1	79. 2	
0. 17	85. 9	0. 17	85. 4	79. 5	
0. 18	87. 4	0. 18	85. 5	79. 6	
0. 19	88. 8	0. 19	85. 5	79. 6	
0. 2	90. 2	0. 2	85. 4	79. 5	
0. 21	91. 5	0. 21	85. 2		
0. 22	92. 8	0. 22	84. 9		
0. 23	94. 1	0. 23	84. 5		
0. 24	95. 3	0. 24	84		
0. 25	96. 4	0. 25	83. 4		
0. 26	97. 6	0. 26	82. 8		
0. 27	98. 7	0. 27	82		
0. 28	99. 8	0. 28	81. 3		
0. 29	100. 8	0. 29	80. 4		
0. 3	101. 9	0. 3	79. 6		
0. 31	102. 9				
0. 32	103. 9				

(continued)

$\frac{D_w^{①}}{D_{pw}}$	f_e	$\frac{D_w \cos \alpha^{①}}{D_{pw}}$	f_c		
	$\alpha = 90^\circ$		$\alpha = 45^\circ$ ②	$\alpha = 60^\circ$	$\alpha = 75^\circ$
0. 33	104. 8				
0. 34	105. 8				
0. 35	106. 7				

Note: ① Values of f_c for $\frac{D_w}{D_{pw}}$ or $\frac{D_w \cos \alpha}{D_{pw}}$ and/or contact angles other than those shown in the table are obtained by linear interpolation.

② For thrust bearings $\alpha > 45^\circ$. Values for $\alpha = 45^\circ$ are given to permit interpolation of values for α between 45° and 60° .

Table 2 - 35 Values of X and Y for thrust ball bearings

$a^{①}$	Single bearing ^②		Double direction bearing				e
	$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		
	X	Y	X	Y	X	Y	
45° ^③	0. 66	1	1. 18	0. 59	0. 66	1	1. 25
50°	0. 73		1. 37	0. 57	0. 73		1. 49
55°	0. 81		1. 6	0. 56	0. 81		1. 79
60°	0. 92		1. 9	0. 55	0. 92		2. 17
65°	1. 06		2. 3	0. 54	1. 06		2. 68
70°	1. 28		2. 9	0. 53	1. 28		3. 43
75°	1. 66		3. 89	0. 52	1. 66		4. 67
80°	2. 43		5. 86	0. 52	2. 43		7. 09
85°	4. 8		11. 75	0. 51	4. 8		14. 29
$\alpha \neq 90^\circ$	$1.25 \tan \alpha * \left(1 - \frac{2}{3} \sin \alpha\right)$	1	$\frac{20}{13} \tan \alpha * \left(1 - \frac{1}{3} \sin \alpha\right)$	$\frac{10}{13} \left(1 - \frac{1}{3} \sin \alpha\right)$	$1.25 \tan \alpha * \left(1 - \frac{2}{3} \sin \alpha\right)$	1	$1.25 \tan \alpha$

Note: ① Values of X , Y and e for intermediate values of α are obtained by linear interpolation.

② $F_a/F_r \leq e$ is unsuitable for single-direction bearings.

③ For $\alpha > 45^\circ$ thrust bearings, values for $\alpha = 45^\circ$ are given to permit interpolation of values for α between 45° and 50° .

2.7.3 Radial Roller Bearings

1. Basic dynamic radial load rating

The basic dynamic radial load rating C_r for a radial roller bearing is given by:

$$C_r = b_m f_c (i L_{we} \cos \alpha)^{7/9} Z^{3/4} D_{we}^{29/27}$$

Where the values of b_m and f_c are given in Table 2 – 36 and 2 – 37, respectively. The listed values in the tables are the maximum values applicable only to roller bearings in which, under bearing loads, the contact stress is substantially uniform along the most heavily loaded roller/raceway contact.

Values of f_c smaller than those given in Table 2 – 37 should be used if, under loads, accentuated stress concentration is present in some part of the roller/raceway contact. Such stress concentration are to be expected, at the center of the nominal contact points, at the extremities of the line contact, in bearings where the rollers are not accurately guided and in bearings having rollers longer than 2.5 times their diameter.

Table 2 – 36 Values of b_m for radial roller bearings

Bearing type	b_m
cylindrical roller bearings, tapered roller bearings and	
needle roller bearings with machined rings	1.1
drawn cup needle roller bearings	1
spherical roller bearings	1.15

Table 2 – 37 Maximum values of f_c for radial roller bearings

$\frac{D_{we} \cos \alpha^{①}}{D_{pw}}$	f_c	$\frac{D_{we} \cos \alpha^{①}}{D_{pw}}$	f_c
0.01	52.1	0.16	88.5
0.02	60.8	0.17	88.7
0.03	66.5	0.18	88.8
0.04	70.7	0.19	88.8
0.05	74.1	0.2	88.7

(continued)

$\frac{D_{we} \cos \alpha^{①}}{D_{pw}}$	f_c	$\frac{D_{we} \cos \alpha^{①}}{D_{pw}}$	f_c
0.06	76.9	0.21	88.5
0.07	79.2	0.22	88.2
0.08	81.2	0.23	87.9
0.09	82.8	0.24	87.5
0.1	84.2	0.25	87
0.11	85.4	0.26	86.4
0.12	86.4	0.27	85.8
0.13	87.1	0.28	85.2
0.14	87.7	0.29	84.5
0.15	88.2	0.3	83.8

① Values of f_c for intermediate values of $\frac{D_{we} \cos \alpha}{D_{pw}}$ are obtained by linear interpolation.

2. Basic dynamic radial load rating for bearing combinations

When calculating the basic dynamic load rating for two identical single-row radial roller bearings mounted side by side on the same shaft, operated as a unit (paired mounting) in a back-to-back or face-to-face arrangement, the pair is consider as one double-row bearing.

If for some technical reasons, the bearing arrangement is regarded as two bearings which are replaceable independently of each other, then the above stipulation does not apply.

The basic dynamic radial load rating, for two or more identical single-row radial roller bearings mounted side by side on the same shaft, operated as a unit (paired or stack mounting) in a tandem arrangement, is the number of pairs of bearings to the power of 7/9 times the basic dynamic radial load rating of one single-row bearing. The proper manufacturing and mounting can ensure even load distribution between the bearings.

If for some technical reasons, the bearing arrangement is regarded as two bearings which are replaceable independently of each other, then the above stipulation does not apply.

3. Dynamic equivalent radial load

The dynamic equivalent radial load for radial roller bearing with $\alpha = 0^\circ$,

under constant radial and axial loads, is given by:

$$P_r = XF_r + YF_a$$

Where the values of X and Y are given in Table 2 – 38.

Table 2 – 38 Radial roller bearing X and Y Values

Bearing type	$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		e
	X	Y	X	Y	
single row $\alpha \neq 0$	1	0	0.4	$0.4 \cot \alpha$	$1.5 \tan \alpha$
double row $\alpha \neq 0$	1	$0.45 \cot \alpha$	0.67	$0.67 \cot \alpha$	$1.5 \tan \alpha$

The radial roller bearings with $\alpha = 0^\circ$ can support radial load only. The dynamic equivalent radial load for this type of bearings is given by:

$$P_r = F_r$$

Note: The ability of radial roller bearings with $\alpha = 0^\circ$ to support axial loads varies considerably with bearing design and manufacturing. The bearing user should therefore consult the bearing manufacturer for recommendations regarding the evaluation of equivalent load and life for cases when bearings with $\alpha = 0^\circ$ are subjected to axial load.

4. Dynamic equivalent radial load for bearing combinations

When calculating the equivalent radial load for two identical single-row angular contact roller bearings mounted side by side on the same shaft, operated as a unit (paired mounting) in a back-to-back or a face-to-face arrangement, the pair is considered as one double-row angular contact roller bearing. The values X and Y for double-row bearing given in Table 2 – 38 shall be used.

When calculating the equivalent radial load for two or more identical radial contact roller bearings or two or more identical angular contact roller bearings mounted side by side on the same shaft, operated as a unit (paired or stack mounting) in a tandem arrangement, the values of X and Y for a single-row bearing given in Table 2 – 38 shall be used.

5. Basic rating life

1) The basic rating life for a radial roller bearing is given by the life equation:

$$L_{10} = \left(\frac{C_r}{P_r} \right)^{10/3}$$

The values of C_r and P_r are calculated in accordance with the above introduction method.

The life equation is also used for the evaluation of the life of two or more single-row bearings, as referred to bearing combinations. In this case, the load rating C_r is calculated for the entire bearing arrangement and the equivalent load P_r is calculated for the total loads acting on the arrangement, using the values of X and Y of double-row bearing indicated in Table 2-38.

2) The life equation gives satisfactory results for a broad range of bearing loads. However, extra-heavy loads may cause accentuated stress concentrations in some part of the roller/raceway contacts. The user should therefore consult the bearing manufacturer to establish the applicability of the life equation when P_r exceeds $0.5C_r$.

2.7.4 Thrust Roller Bearings

1. Basic dynamic axial load rating

(1) Single-row bearings

1) A thrust roller bearing is considered as a single-row bearing only if all rollers carrying load in the same direction contact the same washer raceway area.

The basic dynamic axial load rating C_a for single-row, single-direction or double direction thrust roller bearing is given by:

$$\text{For } \alpha = 90^\circ, \quad C_a = b_m f_c L_{we}^{7/9} Z^{3/4} D_{we}^{29/27}$$

$$\text{For } \alpha \neq 90^\circ, \quad C_a = b_m f_c (L_{we} \cos \alpha)^{7/9} \tan \alpha Z^{3/4} D_{we}^{29/27}$$

Where Z is the number of rollers carrying load in the same direction.

2) If several rollers with their axes coinciding are installed on the same side of the bearing axis, these rollers are considered as one roller with a length L_{we} equal to the sum of the length of these rollers.

Values of b_m and f_c are given in Tables 2-39 and 2-40. The listed values are the maximum values, only applicable to roller bearings in which, under bearing loads, the contact stress is substantially uniform along the most heavily loaded roller/raceway contact.

Values of f_c smaller than those given in Table 2 – 40 should be used if, under loads, an accentuated stress concentration is present in some part of the roller/raceway contact. Such stress concentrations must be expected, for example, at the center of nominal point contacts, at the extremities of line contacts, in bearings where the rollers are not accurately guided, and in bearings with rollers longer than 2.5 times the roller diameter.

Smaller values of f_c should be considered for thrust roller bearings in which the geometry causes excessive slip in the roller/raceway contact areas, for example for the bearings with cylindrical rollers which length in relation to the pitch diameter of the roller set is large.

Table 2 – 39 Values of b_m for thrust roller bearings

Bearing type	b_m
cylindrical roller bearings and needle roller bearings	1
tapered roller bearings	1. 1
spherical roller bearings	1. 15

Table 2 – 40 Maximum values of f_c for thrust roller bearings

$\frac{D_{we}^{①}}{D_{pw}}$	f_c	$\frac{D_{we} \cos \alpha^{①}}{D_{pw}}$	f_c		
	$\alpha = 90^\circ$		$\alpha = 50^\circ$ ②	$\alpha = 65^\circ$ ③	$\alpha = 80^\circ$ ④
0. 01	105. 4	0. 01	109. 7	107. 1	105. 6
0. 02	122. 9	0. 02	127. 8	124. 7	123
0. 03	134. 5	0. 03	139. 5	136. 2	134. 3
0. 04	143. 4	0. 04	148. 3	144. 7	142. 8
0. 05	150. 7	0. 05	155. 2	151. 5	149. 4
0. 06	156. 9	0. 06	160. 9	157	154. 9
0. 07	162. 4	0. 07	165. 6	161. 6	159. 4
0. 08	167. 2	0. 08	169. 5	165. 5	163. 2
0. 09	171. 7	0. 09	172. 8	168. 7	166. 4
0. 1	175. 7	0. 1	175. 5	171. 4	169
0. 11	179. 5	0. 11	177. 8	173. 6	171. 2
0. 12	183	0. 12	179. 7	175. 4	173
0. 13	186. 3	0. 13	181. 1	176. 8	174. 4

(continued)

$\frac{D_{we}^{①}}{D_{pw}}$	f_c	$\frac{D_{we} \cos \alpha^{①}}{D_{pw}}$	f_c		
	$\alpha = 90^\circ$		$\alpha = 50^\circ$ ②	$\alpha = 65^\circ$ ③	$\alpha = 80^\circ$ ④
0. 14	189. 4	0. 14	182. 3	177. 9	175. 5
0. 15	192. 3	0. 15	183. 1	178. 8	176. 3
0. 16	195. 1	0. 16	183. 7	179. 3	
0. 17	197. 7	0. 17	184	179. 6	
0. 18	200. 3	0. 18	184. 1	179. 7	
0. 19	202. 7	0. 19	184	179. 6	
0. 2	205	0. 2	183. 7	179. 3	
0. 21	207. 2	0. 21	183. 2		
0. 22	209. 4	0. 22	182. 6		
0. 23	211. 5	0. 23	181. 8		
0. 24	213. 5	0. 24	180. 9		
0. 25	215. 4	0. 25	179. 8		
0. 26	217. 3	0. 26	178. 7		
0. 27	219. 1				
0. 28	220. 9				
0. 29	222. 7				
0. 3	224. 3				

Note: ① Values of f_c for intermediate values of $\frac{D_{we}}{D_{pw}}$ or $\frac{D_{we} \cos \alpha}{D_{pw}}$ are obtained by linear interpolation.

- ② Applicable for $45^\circ < \alpha < 60^\circ$.
- ③ Applicable for $60^\circ \leq \alpha < 75^\circ$.
- ④ Applicable for $75^\circ \leq \alpha < 90^\circ$.

(2) Basic dynamic axial load rating for bearings with two or more rows of rollers The basic dynamic axial load rating for bearings with two or more rows of rollers carrying load in the same direction is given by:

$$C_a = (Z_1 L_{we1} + Z_2 L_{we2} + \dots + Z_n L_{wen}) \times \left[\left(\frac{Z_1 L_{we1}}{C_{a1}} \right)^{9/2} + \left(\frac{Z_2 L_{we2}}{C_{a2}} \right)^{9/2} + \dots + \left(\frac{Z_n L_{wen}}{C_{an}} \right)^{9/2} \right]^{-2/9}$$

The load rating C_{a1} , C_{a2} , ..., C_{an} for the rows with Z_1 , Z_2 , Z_3 , ..., Z_n rollers of length L_{we1} , L_{we2} , ..., L_{wen} are calculated from the appropriate

single-row bearing equations.

Rollers and/or portions of rollers which contact the same washer raceway are belong to one row.

(3) Basic dynamic axial load rating for bearing combinations

The basic dynamic radial load rating, for two or more identical single-direction thrust roller bearings mounted side by side on the same shaft, operated as a unit (paired or stack mounting) in a tandem arrangement, is the number of pairs of bearings to the power of 7/9 times the rating of one bearing. The proper manufacturing and mounting can ensure even load distribution between the bearings.

If for some technical reasons, the bearing arrangement is regarded as a number of single-direction bearings which are replaceable independently of each other, then the above stipulation does not apply.

2. Dynamic equivalent axial load

The dynamic equivalent axial load for thrust roller bearing with $\alpha \neq 90^\circ$, under constant radial and axial loads, is given by:

$$P_a = XF_r + YF_a$$

Where the values of X and Y are given in Table 2-41.

Thrust roller bearing with $\alpha = 90^\circ$ can support axial load only. The dynamic equivalent axial load for this type of bearing is given by:

$$P_a = F_a$$

Table 2-41 Values of X and Y for thrust roller bearing

Bearing type	$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		e
	X	Y	X	Y	
single-direction, $\alpha \neq 90^\circ$	— ^①	— ^①	$\tan \alpha$	1	$1.5 \tan \alpha$
double-direction, $\alpha \neq 90^\circ$	$1.5 \tan \alpha$	0.67	$\tan \alpha$	1	$1.5 \tan \alpha$

① Unsuitable for single-direction bearings.

3. Basic rating life

1) The basic rating life for a thrust roller bearing is given by

$$L_{10} = \left(\frac{C_a}{P_a} \right)^{10/3}$$

The values of C_a and P_a are calculated in accordance with the above

introduction method.

This life equation is also used for the evaluation of two or more single-direction thrust roller bearings operated as a unit, as referred in the above mentioned for bearing combinations.

In this case the load rating C_a is calculate for the entire bearing arrangement and the equivalent load P_a is calculated for the tatol loads acting on the arrangement, using the values of X and Y given for single-direction bearing in Table 2 – 41.

2) The life equation gives satisfactory results for a broad range of bearing loads. However, extra-heavy loads may cause accentuated stress concentration in some part of the roller/raceway contacts. The user should therefore consult the bearing manufacturer to establish the applicability of the life equation in case when P_a exceeds $0.5C_a$.

2.7.5 Modified Rating Life

1. Generals

For many years, the use of the basic rating life L_{10} as a criterion of bearing performance has shown satisfactory. This life is associated with 90% reliability, with commonly used high quality material, good manufacturing quality, and with conventional operating conditions.

However, for many applications it has become desirable to calculate the life for a different level of reliability and/or for more accurate life calculation under specified lubrication and contamination conditions. With modern high quality bearing steel, it has been found that, under favorable operating conditions and below certain rolling element contact stress, very long bearing lives compared with L_{10} life, can be obtained if the fatigue limit of the bearing steel is not exceeded. On the other hand, bearing lives shorter than the L_{10} life can be obtained under unfavorable operating conditions.

A system approach to the fatigue life calculation has been used in this international standard. With such a method, the influence on the life of the system due to variation and interaction of interdependent factors is considered by referring all influences to the additional stress they give rise to the rolling element contacts and the contact regions.

In this international standard, a life modification factor a_{ISO} , is introduced, based on a system approach of life calculation in addition to the modification factor a_1 , These factors are applied in the modified rating life equation given as follows:

$$L_{\text{nm}} = a_1 a_{\text{ISO}} L_{10}$$

The modification factor for reliability, a_1 , for a range of reliability values is given in Table 2 – 42, and the method for evaluating the modification factor for a system approach, a_{ISO} , is detailed in Section 2. 7. 5. 3.

- Note: 1. In ISO 281—1990, the modification factor is a_{xyz} , in ISO 281—2007, modified it as a_{ISO} .
2. In L_{nm} , the $n\%$ is the probability of failure, the $(100 - n)\%$ is the probability of survival, that is the reliability.

Table 2 – 42 Life modification factor for reliability, a_1

Reliability	L_{nm}	a_1
90	$L_{10\text{m}}$	1
95	$L_{5\text{m}}$	0. 62
96	$L_{4\text{m}}$	0. 53
97	$L_{3\text{m}}$	0. 44
98	$L_{2\text{m}}$	0. 33
99	$L_{1\text{m}}$	0. 21
99. 2	$L_{0. 8\text{m}}$	0. 22
99. 4	$L_{0. 6\text{m}}$	0. 19
99. 6	$L_{0. 4\text{m}}$	0. 16
99. 8	$L_{0. 2\text{m}}$	0. 12
99. 9	$L_{0. 1\text{m}}$	0. 093
99. 92	$L_{0. 08\text{m}}$	0. 087
99. 94	$L_{0. 06\text{m}}$	0. 080
99. 95	$L_{0. 05\text{m}}$	0. 077

2. Life modification factor for reliability a_1

In Table 2 – 42, the values of reliability can be calculated by the following equation (in ISO 281—1990). But in ISO 281—2007, the a_1 values for the reliability 95% to 99% have been modified slightly compared

with the corresponding values in the previous edition of this international standard.

$$a_1 = \left[\frac{\ln \frac{100}{s}}{\ln \frac{100}{90}} \right]^{\frac{1}{e}}$$

In the equation, e is the Weibull exponent. Table 2 – 42 takes $e = 1.5$. We also can calculate a_1 value by other e values.

3. Life modification factor for system approach a_{ISO}

Below a certain load, a modern high quality bearing can attain an infinite life, if the lubrication conditions, the cleanliness and other operation conditions are favorable.

For rolling bearings of commonly used high quality material and good manufacturing quality, the fatigue stress limit is reached at a contact stress of approximately 1500MPa. This stress value takes into account additional stresses occurring due to manufacturing tolerances and operating conditions. Reduced manufacturing accuracy and/or material quality result in a lower fatigue stress limit.

In many applications, contact stress are, however, larger than 1500MPa and, in addition, the operating conditions can give rise to additional stresses and by that further reduce the bearing life.

It is possible to relate all operating influences to the applied stresses and to the strength of the material, e. g. :

- indentations give rise to edge stresses;
- a thin oil film increases the stresses in the contact region between raceway and rolling element;
- an increased temperature reduces the fatigue stress limit of the material, i. e. its strength;
- a tight inner ring fit gives rise to hoop stresses.

The different influences on bearing life are dependent on each other. A system approach of the fatigue life calculation is therefore appropriate, as the influence on the life of the system from variation and interaction of interdependent factors will then be considered. For performing modified system approach calculations, practical methods have been developed for

determining the life modification factor, a_{ISO} , which consider the fatigue stress limit of the bearing steel and make it easy to estimate the influence of lubrication and contamination on bearing life.

The life modification factor for system approach, a_{ISO} , can be expressed as a function of σ_u/σ , the fatigue stress limit divided by the real stress with as many influencing factors as possible considered, as given in Figure 2-49.

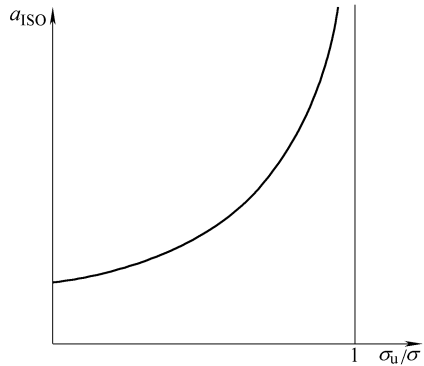


Figure 2-49 Life modification factor for system approach a_{ISO}

2.8 Plastic Deformation in Rolling Bearings

2.8.1 Palmgren Theory on Rolling Bearing Static Load Capacity

Figure 2-50 is a sketch of plastic deformation. Under an applied load Q , for the three identical diameter balls, the central distance of ball I and III reduced to l_1 from l , the point a on the ball I and the point b on ball II, the point c on the ball II and the point d on the ball III approached to each other. The total approach amount is:

$$\delta = l - l_1$$

If the approach δ is due to complete elastic deformation, then their central distance would fully recover to l when the load is removed. But actually it is just recovered to l_2 . That is to express that the contact deformations between the ball I and the ball II and between the ball II and the ball III are not complete elastic deformations, but there is an existence of plastic deformations.

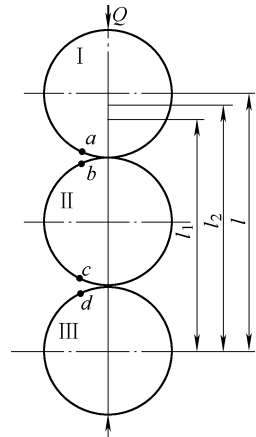


Figure 2-50 Plastic deformation

The plastic deformation is:

$$\delta_s = l - l_2$$

According to the Hertz elastic theory, for point contact, the elastic deformation is proportional to the power $2/3$ of load Q , in Figure 2-51, the elastic deformation should be a straight line expressed by the dotted line. But the actually measured deformation curve is a curve line. Figure 2-51 shows that when the load is small, the contact deformation is mainly elastic deformation. With the load increasing, the plastic deformation also increases. If the load exceeds some value, the plastic deformation also obviously increases. Figure 2-52 shows the relationship of the plastic deformation δ_s with the load Q .

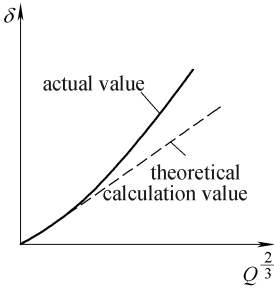


Figure 2-51 δ and $Q^{2/3}$

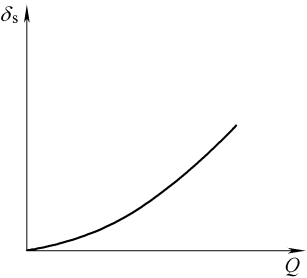


Figure 2-52 Plastic deformation δ_s with the load Q

The form of plastic deformation in rolling bearings is forming a pit on the contact surface. Experiment has proved that, in practice, even under very light load, plastic deformations of small magnitude also occur. The main reason for producing plastic deformation under very light load is that the contact surfaces are not ideally smooth, but are rough and uneven. While two bodies entering into contact each other, first of all, the first entering contact points are the jut points of convex. Because the area of the jut point is very small, even though the load is very light, the pressure per area has exceeded the elastic limit of the material, therefore, the jut points are subjected to plastic deformation and gradually becoming flat.

2. Plastic deformation calculation equation

On the basis of empirical data for bearing quality steel, Palmgren developed the following formula to describe plastic (permanent) deformation:

For point contact:

$$\delta_s = 1.29 * 10^{-7} \frac{Q^2}{D_w} (\rho_{I1} + \rho_{II1}) (\rho_{I2} + \rho_{II2}) \quad (2-316)$$

Where δ_s — total plastic deformation amount at one rolling element/raceway contact place (mm);

Q — rolling element load (N);

D_w — rolling element diameter (mm);

$\rho_{I1}, \rho_{II1}, \rho_{I2}, \rho_{II2}$ — the main curvatures of body I and body II in principle plane 1 and principle plane 2 (1/mm).

For line contact:

Under line contact condition, plastic (permanent) deformation amount is different along the roller length. On the middle part of the roller, the plastic (permanent) deformation amount is smaller. On the contacting end part of the roller, the plastic (permanent) deformation amount is larger, especially for the condition where the ring raceway length is larger than the roller length.

The maximum total plastic (permanent) deformation amount is given by:

$$\delta_s = \frac{2.12 * 10^{-11}}{\sqrt{D_w}} \left(\frac{Q}{l_{we}} \sqrt{\rho_I + \rho_{II}} \right)^3 \quad (2-317)$$

Where δ_s — maximum total plastic deformation amount on roller (mm);

l_{we} — rolling element effective contact length (mm);

ρ_I, ρ_{II} — the main curvatures of body I and body II.

2.8.2 Rolling Bearing Static Load Constant

From the formula of plastic deformations (2-316) and (2-317) we can derive the following equations:

For point contact:

$$k = \frac{Q}{D_w^2} = \frac{2784}{D_w \sqrt{(\rho_{I1} + \rho_{II1}) (\rho_{I2} + \rho_{II2})}} \left(\frac{\delta_s}{D_w} \right)^{\frac{1}{2}} \quad (2-318)$$

For line contact:

$$k = \frac{Q}{D_w l_{wc}} = \frac{3613}{\sqrt{D_w} (\rho_I + \rho_{II})} \left(\frac{\delta_s}{D_w} \right)^{\frac{1}{3}} \quad (2-319)$$

From the equations (2-318) and (2-319) we can see that if the δ_s/D_w is a certain value, then the k value is a constant. Generally speaking, given a permissible plastic deformation amount of δ_s/D_w value, then from the equations (2-318) and (2-319) we can decide the permissible static load constant and then can decide the permissible rolling element load from the static load constant k .

From the equations (2-318) and (2-319) we can get:

For point contact:

$$Q = k D_w^2 \quad (2-320)$$

For line contact:

$$Q = k D_w l_{wc} \quad (2-321)$$

When determining the bearing static load rating, the stipulated plastic deformation amount is as: $\delta_s/D_w = 0.0001$, if using k_0 expresses the static load constant, then from the equations (2-318) and (2-319) we can get:

For ball bearing:

$$k_0 = \frac{27.84}{D_w \sqrt{(\rho_{I1} + \rho_{II1}) (\rho_{I2} + \rho_{II2})}} \quad (2-322)$$

For roller bearing:

$$k_0 = \frac{167.7}{\sqrt{D_w} (\rho_I + \rho_{II})} \quad (2-323)$$

From the equations (2-322) and (2-323) we can see that the right side of these equations is a constant relating to the bearing structure, therefore, for a given rolling bearing, we can calculate the static load constant k_0 . For the same type bearings, the k_0 values changes within certain range. For simplified calculation, for standard design of the same type bearings, we will take an average value. Different type bearing k_0 values are given in Table 2-43.

Table 2 – 43 Standard design bearing static load constant k_0

Bearing type	k_0	Bearing type	k_0
radial ball bearing	60.8	radial cylindrical roller bearing	107.8
angular contact bearing	60.8	self-aligning roller bearing	107.8
separable angular contact ball bearing	14.7	tapered roller bearing	107.8
self-aligning ball bearing	16.7	thrust roller bearing	98.1
thrust ball bearing	49		

2.8.3 Basic Static Load Rating

1. Definition

The load capacity of stationary rolling bearing is decided by the permanent deformation amount. Some degree of permanent deformation is unavoidable in loaded rolling bearings. If the permissible permanent deformation amount is very small, then the static load capacity is also very small; If the permissible permanent deformation amount is large, then the cavities formed in the raceways cause the bearing to vibrate and the bearing becomes noisy, and the running accuracy will be reduced, therefore, bearing normal working will be affected.

A long time rolling bearing application experience has shown that the total plastic (permanent) deformations (the raceway plastic deformation + rolling element plastic deformation) have little effect on the operation of the bearing if the magnitude at any given contact point is limited to a maximum of $0.0001D_w$. The rolling bearing static load rating is determined on this basis, and is expressed by C_0 .

The rolling bearing static load rating is determined on the basis of the assumed load condition. That is for radial bearing, static load rating designates the radial load; for angular contact bearing designates the radial element of a load that under this load the bearing is just half-raceway loading; for thrust bearing designates the central axial load.

2. Determining method for static load rating

The steps for determining the static load rating are as follows:

1) According to the bearing structure, we determine at which rolling

element/raceway contact point the plastic deformation amount is the largest under the same actual load condition. A large plastic deformation amount of the contact point means that the strength is weak in this contact point. Therefore, using the structure dimensions of this contact point calculates the static load constant;

2) From the equations (2 - 322) and (2 - 323), we calculate the static load constant for $\delta_s = 0.0001D_w$;

3) From the equations (2 - 320) and (2 - 321), we calculate the permissible maximum $Q_{\max}$ value in the bearing;

4) From the bearing load distribution relationship and the permissible maximum $Q_{\max}$ value in the bearing, we calculate the static load rating C_0 .

3. Static load rating C_0 equations

For standard design bearing, we can check out the k_0 value from Table 2 - 43, then getting the static load rating calculation equations from the bearing load distribution relationship.

(1) Radial contact ball bearing and angular contact ball bearing

In the bearing load distribution equation (2 - 78), if the load distribution factor $T = 0.5$, and $Q_{\max}$ is the maximum load acting on the rolling element causing a plastic deformation amount of $\delta_s = 0.0001D_w$, then F_r is the bearing static load rating C_0 . From the equations (2 - 78) and (2 - 320) we can get:

$$C_0 = k_0 J_r (0.5) z D_w^2 \cos \alpha$$

For i -row bearing, the general expression of bearing static load rating is:

$$C_0 = f_0 i z D_w^2 \cos \alpha \quad (2 - 324)$$

$$f_0 = k_0 J_r (0.5) \quad (2 - 325)$$

Where C_0 — static load rating (N);

f_0 — coefficient, relating to the bearing type and the radial integral of load distribution. Different bearing type f_0 are given in Table 2 - 44;

i — rolling element rows;

z — rolling element number per row;

α — contact angle;

D_w — rolling element diameter (mm).

Table 2 – 44 Different type bearing f_0

Bearing type	f_0	Bearing type	f_0
radial ball bearing	12.3	radial cylindrical roller bearing	21.6
angular contact bearing	12.3	self-aligning roller bearing	21.6
separable angular contact ball bearing	2.94	tapered roller bearing	21.6
self-aligning ball bearing	3.33	thrust roller bearing	98.1
thrust ball bearing	49		

(2) Thrust ball bearing

For thrust and thrust angular contact bearing under central axial load, every rolling element subjects to the same load:

$$Q = \frac{F_a}{z \sin \alpha}$$

If Q is the rolling element load of the plastic deformation amount of $\delta_s / D_w = 0.0001$, then F_a is the thrust bearing static load rating C_0 . From the equation (2 – 320) we can get:

$$C_0 = f_0 z D_w^2 \sin \alpha \quad (2 - 326)$$

Where

$$f_0 = k_0$$

(3) Radial roller bearings

From the equations (2 – 78) and (2 – 321) we can get radial roller bearing static load rating as:

$$C_0 = f_0 i z D_w l_{we} \cos \alpha \quad (2 - 327)$$

$$\text{Where } f_0 = k_0 J_r \quad (0.5) \quad (2 - 325)$$

l_{we} — effective contact length (mm).

(4) Thrust roller bearings

Using the same deriving method with the thrust ball bearing we can get:

$$C_0 = f_0 z D_w l_{we} \sin \alpha \quad (2 - 328)$$

2.8.4 Equivalent Static Load

The rolling bearing static load rating is determined on the basis of the assumed load condition. For radial bearings and angular contact bearings, we assume that there is only a relative radial displacement for the inner ring

and outer ring, that is the load distribution factor $T = 0.5$. For thrust bearings and thrust angular contact bearings, there is only a relative axial displacement, that is the load distribution factor $T = \infty$. If the actual bearing load condition is different with the assumed condition, first we convert the actual load into the static equivalent load and then we can compare the static equivalent load with the static load rating.

The static equivalent load is an assumed load. Under that load, the total plastic deformation due to the maximum rolling element load at the contact point of rolling element/raceway is the same as the total plastic deformation of the actual load. For radial bearings, the static equivalent load is a radial load; for thrust and thrust angular contact bearings, the static equivalent load is a central axial load; for angular contact bearings, the static equivalent load is the radial load (component) of the load, under that load there are only half-raceways loading.

1. The method for determining static equivalent load

From the load distribution equation, we can get the maximum rolling element load under any load as:

$$Q_{\max} = \frac{F_r}{J_r(T) z \cos \alpha} = \frac{F_a}{z J_a(T) \sin \alpha} \quad (2-84)$$

From the definition of the static equivalent load we can get:

$$Q_{\max} = \frac{P_0}{J_r(0.5) z \cos \alpha} \quad (2-329)$$

From the equations (2-328) and (2-329), we can get:

$$\frac{F_r}{P_0} = \frac{J_r(T)}{J_r(0.5)} \quad (2-330)$$

$$\frac{F_a \cot \alpha}{P_0} = \frac{J_a(T)}{J_r(0.5)} \quad (2-331)$$

Where $Q_{\max}$ — maximum rolling element load in bearing;

F_r — actual radial load acting on bearing;

$J_r(T)$ — load distribution radial integral;

$J_a(T)$ — load distribution axial integral;

T — load distribution factor;

z — rolling element number;

α — contact angle;

P_0 — static equivalent load;

$J_r (0.5)$ — load distribution radial integral for half-raceway loading.

For different load distribution factor T , we can calculate the corresponding F_r/P_0 and $F_a \cot \alpha / P_0$. Figure 2-53 and Figure 2-54 show the relationships of point contact and line contact for F_r/P_0 and $F_a \cot \alpha / P_0$. The curve lines show the relationships of F_r and F_a for getting the same static equivalent load P_0 .

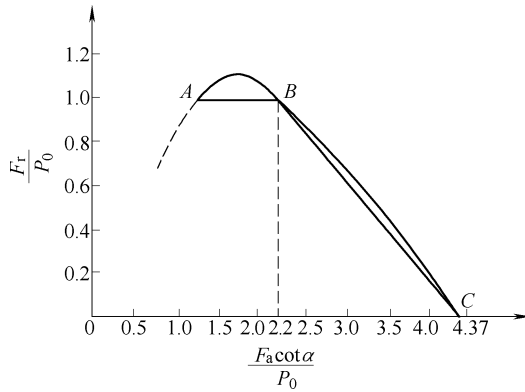


Figure 2-53 Point contact

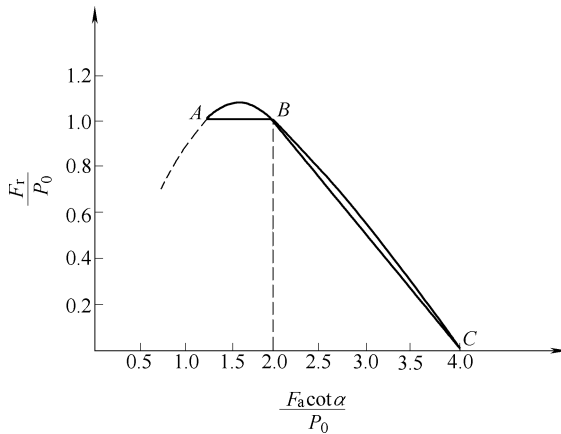


Figure 2-54 line contact

In Figures (2-53) and (2-54), if using two straight lines of AB, BC instead of the curve lines, then we can simplify the static equivalent load

calculation. In the figures, the point A expresses the load condition of half-raceway loading, that is:

For point contact: $F_a = 1.216 F_r \tan \alpha$

For line contact: $F_a = 1.266 F_r \tan \alpha$

The left part of the point A in the figure is the area of load distribution factor $T < 0.5$, because the loading rolling element number is very little in this condition, so, the bearing can not normally work, therefore it is expressed by dotted line.

The equation of the AB part of the straight line is:

$$P_0 = F_r \quad (2-332)$$

The equation of the BC part of the straight line is^①:

$$P_0 = X_0 F_r + Y_0 F_a \quad (2-333)$$

Where X_0 — static radial factor;

Y_0 — static axial factor;

F_r — radial load (N);

F_a — axial load (N);

P_0 — static equivalent load.

Generally speaking, it is first to calculate the P_0 value from the equation (2-333) and then to compare the P_0 value with F_r , if $P_0 < F_r$, then taking: $P_0 = F_r$.

Concerning the values of X_0 and Y_0 , it is better to determine by the coordinates of two special points of the straight line BC ^②. In ISO/R76, it is

Note: ① When calculating the static equivalent load, we can use the judging factor e_0 as in calculating the dynamic equivalent load:

when $F_a/F_r < e_0$, $P_0 = F_r$.

when $F_a/F_r > e_0$, $P_0 = X_0 F_r + Y_0 F_a$

For ball bearing: $e_0 = 2.2 \tan \alpha$

For roller bearing: $e_0 = 2.0 \cot \alpha$

② For point contact, the coordinates of the point B in Figure 2-53 are:

$$F_r/P_0 = 1 \quad F_a \cot \alpha / P_0 = 2.2$$

From the equation (2-333) we can get: $X_0 = 1 - Y_0 F_a / P_0$

From the coordinates of the point C we can get: $Y_0 = 0.2288 \cot \alpha$

Therefore we can get: $X_0 = 1 - 0.2288 \cot \alpha$

For line contact, from Figure 2-54 we can get:

$$Y_0 = 0.2453 \cot \alpha \quad X_0 = 0.505$$

recommended to use the same equations for ball bearing and roller bearing:

$$X_0 = 0.5$$

$$Y_0 = 0.22 \cot \alpha$$

2. Static equivalent load of radial ball bearing and angular contact ball bearing

The contact angle of radial ball bearing and angular contact ball bearing changes with the applied load. Especially while the nominal contact angle is very small or the bearing carries pure axial load, the contact angle changing is more notable. When determining the static equivalent load, it should be the first to calculate the actual contact angle, and then to determine the static equivalent load value.

The values of X_0 and Y_0 of radial ball bearing and angular contact ball bearing are given in Table 2 – 45.

The values listed in Table 2 – 45 are determined on the basis of certain special value of F_a/C_0 . In order to calculate approximately, we can use them as the average value of that type bearing. Actually, for different value of F_a/C_0 , there should be different value. Such as for radial ball bearing, we can have the relationship in Table 2 – 46.

Table 2 – 45 Ball bearing X_0 , Y_0

Bearing type		Single-row bearing		Double-row bearing ^③	
		X_0	Y_0	X_0	Y_0
radial ball bearing ^①		0.6	0.5	0.6	0.5
angular contact ball bearing ^②	$\alpha = 20^\circ$	0.5	0.42	1	0.84
	$\alpha = 25^\circ$	0.5	0.38	1	0.76
	$\alpha = 30^\circ$	0.5	0.33	1	0.66
	$\alpha = 35^\circ$	0.5	0.29	1	0.58
	$\alpha = 40^\circ$	0.5	0.26	1	0.52
self-aligning ball bearing		0.5	$0.22 \cot \alpha$	1	$0.44 \cot \alpha$

Note: ① The maximum permissible vale of F_a/C_0 for radial ball bearing is related to the bearing structure (raceway depth and internal clearance).

② For back-to-back (outer ring wide face to face) or face-to-face (outer ring narrow face-face) pair arrangement of two identical single row angular contact radial ball

bearings, we can take the double row angular contact radial ball bearing values of X_0 , Y_0 . For tandem pair arrangement (outer ring wide and narrow face-to-face) of two or more identical single row angular contact radial ball bearing we can take the value of X_0 , Y_0 .

③ Assuming that the double-row bearing is symmetrical.

Table 2 - 46 Radial ball bearing X_0 , Y_0

$\frac{F_a}{C_0}$	0.025	0.04	0.07	0.13	0.25	0.50	1.0
X_0	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Y_0	1.2	1.1	0.9	0.8	0.7	0.6	0.5

3. Double-row bearing static equivalent load

For double-row bearing, it is assumed that the bearing is symmetrical. It means that both of the contact angle are α and the two rows are expressed by corner note I and II respectively. It assumes that the row I is subjecting to heavier load, that is the plastic deformation is larger on the row I. Therefore, only using $Q_{\max I}$ calculates the bearing static equivalent load.

$$Q_{\max I} = \frac{F_r}{z \cos \alpha J_r (T_I T_{II})} = \frac{F_a}{z \sin \alpha J_a (T_I T_{II})} \quad (2-102)$$

According to the definition of static equivalent load we can get:

$$Q_{\max I} = \frac{P_0}{J_r (0.5, 0.5) z \cos \alpha} \quad (2-334)$$

If using the P_{0I} expresses the single row bearing static equivalent load, according to the definition of static equivalent load we can get:

$$P_0 = 2 P_{0I} \quad (2-335)$$

From the equations (2-102), (2-334), (2-335) we can get:

$$\frac{F_r}{P_{0I}} = \frac{2 J_r (T_I T_{II})}{J_r (0.5, 0.5)} \quad (2-336)$$

$$\frac{F_a \cot \alpha}{P_{0I}} = \frac{2 J_a (T_I T_{II})}{J_r (0.5, 0.5)} \quad (2-337)$$

Figure 2-55 shows the static equivalent load curve line of double-row bearing for point contact, that is the relationship curve line of F_r and F_a in order to get the same static equivalent load P_0 .

For simplified calculation, in Figure 2-55, we use the straight line AC instead of the curve line ABC. The straight line AC equation is:

$$P_0 = X_0 F_r + Y_0 F_a \quad (2-333)$$

If using Y_{0l} expresses the single row bearing static axial factor, then when $F_r = 0$, the row I of the bearing is subjected to the total axial load, at that time we can get:

$$P_{0l} = Y_{0l} F_a \quad (2-338)$$

From the equation (2-333) we can get when $F_r = 0$:

$$P_0 = Y_0 F_a \quad (2-339)$$

Putting the equation (2-335) and Y_{0l} value into the equation (2-339) we can get:

$$Y_0 = 2 Y_{0l} = 0.44 \cot \alpha \quad (2-340)$$

when $F_a = 0$, from Figure 2-55 we can get:

$$F_r / P_{0l} = 2$$

From the equation (2-333) we can get:

$$P_0 = X_0 F_r \quad (2-341)$$

Putting the equation (2-335) into the equation (2-341) we can get:

$$X_0 = 1$$

Therefore, for self-aligning ball bearing, self-aligning roller bearing, double-row tapered roller bearing, we can use the same equation:

$$X_0 = 1 \quad (2-342)$$

$$Y_0 = 0.44 \cot \alpha$$

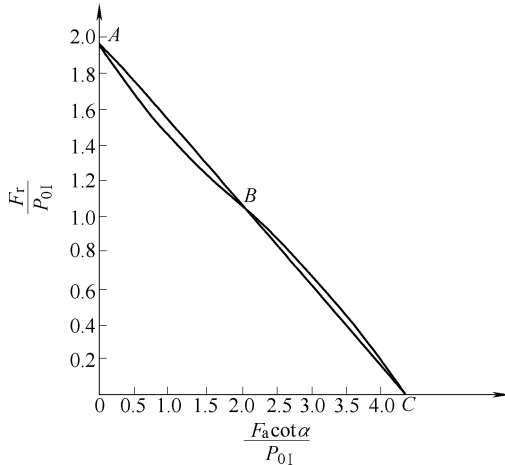


Figure 2-55 Double-row bearing static equivalent load

4. Thrust angular contact bearing static equivalent load

The thrust and thrust angular contact bearing static load rating designates central axial load. Therefore, that type bearing static equivalent load should also be the central axial load. But the thrust angular contact bearing can carry certain radial load, therefore, this type bearing can also be considered as radial angular contact bearing with large contact angle. If using P_{0r} expresses the static equivalent load of half-raceway loading bearing, then from the equation (2-333) we can get:

$$P_{0r} = X_{0r} F_r + Y_{0r} F_a \quad (2-343)$$

Where P_{0r} — static equivalent load expressed by radial load;

X_{0r} — static radial load factor of static equivalent load expressed by radial load;

Y_{0r} — static axial load factor of static equivalent load expressed by radial load.

If F_a is the static equivalent load P_0 expressed by central axial load in this equation, then we can get:

$$P_{0r} = Y_{0r} P_0 \quad (2-344)$$

Putting the equation (2-343) into the equation (2-344) we can get:

$$P_0 = X_{0r} P_0 / Y_{0r} \quad (2-345)$$

If using the uniform expressing equation:

$$P_0 = X_0 F_r + Y_0 F_a \quad (2-333)$$

Then we can get:

$$\begin{aligned} X_0 &= 2.3 \tan \alpha \\ Y_0 &= 1 \end{aligned} \quad (2-346)$$

Where P_0 — static equivalent load expressed by central axial load;

X_0 — static radial factor of static equivalent load expressed by central axial load;

Y_0 — static axial factor of static equivalent load expressed by central axial load.

For thrust angular contact bearing, the radial load should not be too large and should satisfy that $F_r < 0.67 F_a \cot \alpha$. When $F_a / F_r < 2.3 \tan \alpha$, the accuracy of the equation (2-333) will be lowered.

2.9 International Standard ISO 76 and China National Standard GB/T4662

The Palmgren rolling bearing static load capacity theory introduced in this Section 2.8, is applied to the bearing static load carrying capacity calculation under varied kinds of design parameters, and also it is the base of the International Standard ISO 76. Through many years investigation of Palmgren theory, ISO/TC4 (International Standard Organization fourth committee) published a recommended standard ISO/R76 in 1962, which through over ten years investigation was ascended as the official standard in 1987. Later in 1999, it was also increased an annex for this standard. China accepted it equally as the National Standard GB/T 4662 — 2003.

This standard is applied to the rolling bearings which dimensions are in the range of national standard stipulation and which are made by high quality hardening rolling bearing steels of contemporary era, good manufacturing method and the rolling contact surface shapes conforming with common rolling bearing design.

The terms and definitions used in this standard are as follows:

Static load — load acting on a bearing when the speed of rotation of its rings or washers in relation to each other is zero.

Basic static radial load rating C_{or} — radial load which corresponds to the following calculated contact stress at the center of the most heavily loaded rolling element/raceway contact.

- 4600Mpa for self-aligning ball bearing;
- 4200MPa for all other radial ball bearing types; and
- 4000MPa for all radial roller bearings.

In the case of a single-row angular contact bearing, the radial load rating refers to the radial component of that load which causes a purely radial displacement of the bearing rings in relation to each other.

Note: These contact stresses, under static load, refer to the stresses that cause a total permanent deformation of rolling element and raceway which is approximately 0.0001 of the rolling element diameter.

Basic Static axial load rating C_{oa} — axial central load which corresponds to the following calculated stress at the center of the most heavily loaded rolling element raceway contact.

- 4200MPa for thrust ball bearing;
- 4000MPa for thrust roller element bearings.

Note: These contact stresses refer to the stresses that cause a total permanent deformation of rolling element and raceway which is approximately 0.0001 of the rolling element diameter.

Static equivalent radial load P_{or} — static radial load which would cause the same contact stress at the center of the most heavily loaded rolling element/raceway contact as that which occurs under the actual load conditions.

Static equivalent axial load P_{oa} — static center axial which would cause the same contact stress at the center of the most heavily loaded rolling element/raceway contact as that which occurs under the actual load conditions.

Static safety factor — ratio between the basic static load rating and the static equivalent load, giving a margin of safety against inadmissible permanent deformation on rolling elements and raceways.

Roller diameter D_{we} — Theoretical diameter in a radial plane through the middle of the roller length for a symmetrical roller (calculation of load rating).

Note: 1. For a tapered roller, the applicable diameter is equal to the mean value of the diameters at the imaginary sharp corner at the large end and at the small end of the roller.

2. For an asymmetrical convex roller, the applicable diameter is an approximation of the diameter at the point of contact between the roller and the ribless raceway at zero load.

Effective roller length L_{we} — Theoretical maximum length of contact between a roller and that raceway where the contact is shortest.

Note: This is normally taken to be either the distance between the theoretical sharp corners of the roller minus the roller chamfers, or the raceway width excluding the grinding undercuts, whichever is the smaller.

Nominal contact angle α — angle between a plane perpendicular to a

bearing axis (a radial plane) and the nominal line of action of the resultant of the forces transmitted by a bearing ring or washer to a rolling element.

Note: For bearings with asymmetrical rollers, the nominal contact angle is determined by the contact with ribbles raceway.

Pitch diameter of ball set — diameter of the circle containing the center of the balls in one row in a bearing.

Pitch diameter of a roller set — diameter of the circle intersecting the center of the roller axes at the middle of the rollers in one row in a bearing.

C_{0a} — basic static axial load rating (N) ;

C_{0r} — basic static radial load rating (N) ;

D_{pw} — pitch diameter of ball or roller set (mm) ;

D_w — nominal ball diameter (mm) ;

D_{we} — roller diameter applicable in the calculation of load rating (mm) ;

F_a — bearing axial load (axial component of actual bearing load)
(N) ;

F_r — bearing radial load (radial component of actual bearing load)
(N) ;

f_0 — factor for calculation of basic static load rating ;

i — number of rows of rolling elements ;

L_{we} — effective roller length applicable in calculation of load ratings
(mm) ;

P_{0a} — static equivalent axial load (N) ;

P_{0r} — static equivalent radial axial load (N) ;

S_0 — static safety factor ;

X_0 — static radial load factor ;

Y_0 — static radial load factor ;

Z — number of rolling elements in a single-row bearing ; number of rolling elements per row of a multi-row bearing with the same number of rolling elements per row ;

α — nominal contact angle.

2.9.1 Radial Ball Bearings

1. Basic static radial load rating for single bearing

The basic static radial load rating for radial ball bearings is given by the following equation:

$$C_{0r} = f_0 i Z D_w^2 \cos \alpha$$

Where f_0 values are given in Table 2-47.

The equation applies to bearings with a cross-sectional raceway groove radius not larger than $0.52D_w$ in radial and angular contact ball bearing inner rings and not larger than $0.53D_w$ in radial and angular contact ball bearing outer rings and not larger than $0.53D_w$ in self-aligning ball bearing inner rings.

The load-carrying ability of a bearing is not necessarily increased by the use of a groove radius smaller than the above value, but is reduced by the use of a groove radius larger than those indicated in the previous paragraph. In the latter case, a correspondingly reduced value of f_0 shall be used.

Basic static radial load rating for bearing combinations

The basic static radial load rating for two identical single-row radial contact (groove type) ball bearings mounted side by side on the same shaft, operated as a unit (paired mounting), is twice the basic static radial load rating of one single-row bearing.

The basic dynamic radial load rating for two identical single-row angular contact ball bearings mounted side by side on the same shaft, operated as a unit (paired mounting) in a back-to-back or a face-to-face arrangement, is twice the basic static radial load rating of one single-row bearing.

The basic dynamic radial load rating, for two or more identical radial contact ball bearings or two or more identical angular contact ball bearings mounted side by side on the same shaft, operated as a unit (paired or stack mounting) in a tandem arrangement, is the number of pairs of bearings multiplied by the basic static radial load rating of one single-row bearing.

The proper manufacturing and mounting can ensure even load distribution between them.

Table 2 – 47 Values of factor f_0 for ball bearings

$\frac{D_w \cos \alpha}{D_{pw}}$	Factor f_0		
	radial ball bearing		thrust ball bearing
	radial contact and angular contact groove ball bearing	self-aligning ball bearing	
0	14.7	1.9	61.6
0.01	14.9	2	60.8
0.02	15.1	2	59.9
0.03	15.3	2.1	59.1
0.04	15.5	2.1	58.3
0.05	15.7	2.1	57.5
0.06	15.9	2.2	56.7
0.07	16.1	2.2	55.9
0.08	16.3	2.3	55.1
0.09	16.5	2.3	54.3
0.1	16.4	2.4	53.5
0.11	16.1	2.4	52.7
0.12	15.9	2.4	51.9
0.13	15.6	2.5	51.2
0.14	15.4	2.5	50.4
0.15	15.2	2.6	49.6
0.16	14.9	2.6	48.8
0.17	14.7	2.7	48
0.18	14.4	2.7	47.3
0.19	14.2	2.8	46.5
0.2	14	2.8	45.7
0.21	13.7	2.8	45
0.22	13.5	2.9	44.2
0.23	13.2	2.9	43.5
0.24	13	3	42.7

(continued)

$\frac{D_w \cos \alpha}{D_{pw}}$	Factor f_0		
	radial ball bearing		thrust ball bearing
	radial contact and angular contact groove ball bearing	self-aligning ball bearing	
0.25	12.8	3	41.9
0.26	12.5	3.1	41.2
0.27	12.3	3.1	40.5
0.28	12.1	3.2	39.7
0.29	11.8	3.2	39
0.3	11.6	3.3	38.2
0.31	11.4	3.3	37.5
0.32	11.2	3.4	36.8
0.33	10.9	3.4	36
0.34	10.7	3.5	35.3
0.35	10.5	3.5	34.6
0.36	10.3	3.6	
0.37	10	3.6	
0.38	9.8	3.7	
0.39	9.6	3.8	
0.4	9.4	3.8	

Note: This table is based on the Hertzian point contact equation with a modulus of elasticity of $2.07 \times 10^5 \text{ MPa}$ and a poisson's ratio of 0.3. It is assumed that the load distribution results in a maximum ball load of $5 F_r / Z \cos \alpha$ for radial ball bearings and a maximum ball load of $F_a / Z \sin \alpha$ for thrust ball bearings. Values of factor f_0 for intermediate values of $\frac{D_w \cos \alpha}{D_{pw}}$ can be obtained by linear interpolation.

2. Static equivalent radial load

The static equivalent radial load for radial ball bearing is the greater value of the two values given by the following two equations:

$$P_{0r} = X_0 F_r + Y_0 F_a$$
$$P_{0r} = F_r$$

Where the values of factor X_0 and Y_0 are given in Table 2 – 48. These

factors apply to bearings with cross-sectional groove radii according to the number 1 of the section 2.9.1.

Values of Y_0 for intermediate contact angles, not given in Table 2-48, are obtained by linear interpolation.

Table 2-48 Values of factor X_0 and Y_0 for radial ball bearings

Bearing type		Single-row bearings		Double-row bearings	
		X_0	Y_0	X_0	Y_0
radial contact ball bearing		0.6	0.5	0.6	0.5
angular contact ball bearings	15°	0.5	0.46	1	0.92
	20°	0.5	0.42	1	0.84
	25°	0.5	0.38	1	0.76
	30°	0.5	0.33	1	0.66
	35°	0.5	0.29	1	0.58
	40°	0.5	0.26	1	0.52
	45°	0.5	0.22	1	0.44
self-aligning bearing $\alpha \neq 0$		0.5	$0.22\cot\alpha$	1	$0.44\cot\alpha$

Note: The permissible maximum value of F_a/C_{0r} depends on bearing design (internal clearance and raceway groove depth).

3. Static equivalent radial load for bearing combinations

When calculating the basic static equivalent radial load for two identical single-row angular contact ball bearings mounted side by side on the same shaft, operated as a unit (paired mounting) in a back-to-back or a face-to-face arrangement, the factor X_0 and Y_0 values for a double-row bearing and the F_r and F_a values for the total loads on the arrangement shall be used.

When calculating the basic static equivalent radial load for two or more identical single-row radial contact ball bearings or two or more identical single-row angular contact ball bearings mounted side by side on the same shaft, operated as a unit (paired or stack mounting) in a tandem arrangement, the factor X_0 and Y_0 values for a single-row bearing and the F_r and F_a values for the total loads on the arrangement shall be used.

2.9.2 Thrust ball bearings

1. Basic static axial load rating

The basic static axial load rating for single-direction and double-direction thrust ball bearings is given by the following equation:

$$C_{0a} = f_0 Z D_w^2 \sin \alpha$$

Where the values of f_0 are given in Table 2-47, and Z is the number of balls carrying load in one direction.

The equation applies to bearings with cross-sectional raceway groove radii not larger than $0.54D_w$.

The load-carrying ability of a bearing is not necessarily increased by the use of a smaller groove radius, but is reduced by the use of a larger groove radius. In the latter case, a correspondingly reduced value of f_0 shall be used.

2. Static equivalent axial load

The static equivalent axial load for thrust ball bearings with $\alpha \neq 90^\circ$ is given by the following equation:

$$P_{0a} = 2.3 F_r \tan \alpha + F_a$$

This equation is valid for all ratios of radial load to axial load in the case of double-direction bearings. For single-direction bearings, it is valid when $F_r/F_a \leq 0.44 \cot \alpha$ and gives satisfactory but less conservative values of P_{0a} for F_r/F_a up to $0.67 \cot \alpha$.

Thrust ball bearings with $\alpha = 90^\circ$ can support axial loads only. The static equivalent axial load for this type of bearings is given by the following equation:

$$P_{0a} = F_a$$

2.9.3 Radial Roller Bearings

1. static radial load rating

The basic static axial load rating for single-direction and double-direction thrust ball bearings is given by the following equation:

$$C_{0r} = 44 \left(1 - \frac{D_{we} \cos \alpha}{D_{pw}} \right) i Z L_{we} D_{we} \cos \alpha$$

2. Basic static radial load rating for bearing combinations

The basic static radial load rating for two identical single-row radial roller bearings mounted side by side on the same shaft, operated as a unit (paired mounting) in a back-to-back or a face-to-face arrangement, is twice the basic static radial load rating of one single-row bearing.

The basic static radial load rating, for two or more identical radial roller bearings mounted side by side on same shaft, operated as a unit (paired or stack mounting) in a tandem arrangement, is the number of pairs of bearings multiplied by the basic static radial load rating of one single-row bearing. The proper manufacturing and mounting can ensure even load distribution between them.

3. Static equivalent radial load

The static equivalent radial load for roller bearings with $\alpha \neq 0^\circ$ is the greater value of the two values given by the following two equations:

$$P_{0r} = X_0 F_r + Y_0 F_a$$

$$P_{0r} = F_r$$

Where the values of factor X_0 and Y_0 are given in Table 2-49.

Table 2-49 Values of factor X_0 and Y_0 for radial roller bearings with $\alpha \neq 0^\circ$

Bearing type	X_0	Y_0
single row	0.5	$0.22\cot\alpha$
double row	1	$0.44\cot\alpha$

The static equivalent radial load for radial roller bearings with $\alpha = 0^\circ$, and subjected to radial load only, is given by the followings equation:

$$P_{0r} = F_r$$

Note: The ability of radial roller bearings with $\alpha = 0^\circ$ to support axial loads varies considerably with the design and manufacturing of the bearing. The bearing user should therefore consult the bearing manufacturer for recommendations regarding the evaluation of equivalent load in case where bearings with $\alpha = 0^\circ$ are subjected to axial load.

4. Static equivalent axial load for bearings mounted in a tandem arrangement

When calculating the static equivalent axial load for two identical thrust

roller bearings mounted side by side on the same shaft, operated as a unit (paired or stack mounting) in a tandem arrangement, the F_r and F_a values for the total loads on the arrangement shall be used in the above equation.

2.9.4 Thrust Roller Bearings

1. Basic static axial load rating

(1) Basic static axial load rating for single-direction and double-direction bearings

The Basic static axial load rating for single-direction and double-direction thrust roller bearings is given by the equation:

$$C_{0a} = 220 \left(1 - \frac{D_{we} \cos \alpha}{D_{pw}} \right) Z L_{we} D_{we} \sin \alpha$$

where Z is the number of rollers carrying load in the same direction.

In cases where rollers have different lengths, $Z L_{we}$ is taken as the sum of the lengths, as defined in 3.8, of all the rollers carrying load in the same direction.

Note: Equation is based on the same modulus of elasticity. Poisson's ratio and rolling element load distributions as given in the note to Table 1.

(2) Basic static axial load rating for bearings mounted in a tandem arrangement

The basic static axial load rating for two or more identical single-direction thrust roller bearings mounted side by side on the same shaft, such that operated as a unit (paired or stack mounting) in a tandem arrangement, is the number of bearings multiplied by the basic static axial load rating of one single-direction bearing. The proper manufacturing and mounting can ensure even distribution of the load between them.

2. Static equivalent axial load

(1) Static equivalent axial load for single-direction and double-direction bearings

The static equivalent axial load for roller bearing with $\alpha \neq 90^\circ$ is given by the equation:

$$P_{0a} = 2.3 F_r \tan \alpha + F_a$$

This equation is valid for all ratios of radial load to axial load in the

case of double-direction bearings. For single-direction bearings, it is valid where $F_r/F_a \leq 0.44 \cot \alpha$ and gives satisfactory but less conservative values of P_{0a} for F_r/F_a up to $0.67 \cot \alpha$.

Thrust roller bearings with $\alpha = 90^\circ$ can support axial loads only. The static equivalent axial load for this type of bearing is given by the equation:

$$P_{0a} = F_a$$

(2) static equivalent axial load for bearings mounted in a tandem arrangement

When calculating the static equivalent axial load for two or more identical thrust roller bearings mounted side by side on the same shaft, such that operated as a unit (paired or stack mounting) in a tandem arrangement, the F_r and F_a values for the total loads on the arrangement shall be used in the above equation.

2.9.5 Static Safety Factor

1. Generals

The suitability of a bearing selected for heavily loaded applications should be checked to ensure that the basic static load rating is adequate. This can be determined with the aid of the static safety factor S_0 , which is given by the following equation:

$$S_0 = C_{0r}/P_{0r}$$

$$S_0 = C_{0a}/P_{0a}$$

The above first equation applies to radial bearings and the second equation applies to thrust bearings.

Where the bearing is dynamically loaded and the selection has been made on the basis of life, it is also advisable to check that the basic static load rating is adequate for attaining the performance requirements of the application.

The guideline values of S_0 given in Table 2 – 50 and Table 2 – 51 for various types of operation and application requirements regarding smooth and vibration-free running are applicable to rotating bearings and are based on experience.

Table 2 – 50 Guideline values of static safety factor S_0 for ball bearings

Type of operation	S_0 min
Quiet-running applications: smooth-running, vibration-free, high rotational accuracy	2
Normal-running applications: smooth-running, vibration-free, normal rotational accuracy	1
Applications subjected to shock loads: pronounced shock loads	1.5

Note: Where the magnitude of the load is not known, values of S_0 which are at least 1.5 should be used. If the magnitude of the shock loads is known exactly, small values of S_0 can be used.

Table 2 – 51 Guideline values of static safety factor S_0 for roller bearings

Type of operation	S_0 min
Quiet-running applications: smooth-running, vibration-free, high rotational accuracy	3
Normal-running applications: smooth-running, vibration-free, normal rotational accuracy	1.5
Applications subjected to shock loads: pronounced shock loads	3

For thrust spherical roller bearings, a minimum S_0 of 4 is recommended for all types of operation.

For case-hardened, drawn cup needle roller bearings, a minimum S_0 of 5 is recommended for all types of operation.

Note: Where the magnitude of the load is not known, values of S_0 which are at least 3 should be used. If the magnitude of the shock loads is known exactly, small values of S_0 can be used.

2. informative

(1) General

The factors used for the calculation of basic static load rating C_{0r} and C_{0a} according to the international Standard are slightly different for radial and thrust angular contact ball bearings.

Therefore, there is a discontinuity in the calculation of the static axial

load rating (C_{0a}) when a bearing with the contact angle $\alpha = 45^\circ$ is first regarded as a radial bearing ($C_{0a} = C_{0r}/Y_0$) and then as a thrust bearing.

This annex explains why the load rating factors are different, and shows how the load ratings can be recalculated in order to make accurate comparisons under the same conditions.

(2) Symbols

The same symbols are used as presented in clause 4, but some additional symbols also apply;

C_{0aa} — the adjusted basic static axial load rating for a thrust bearing ($\alpha > 45^\circ$) (N);

C_{0ar} — the adjusted basic static axial load rating for a radial bearing ($\alpha \leq 45^\circ$) (N);

r_e — the cross-sectional raceway groove radius of outer ring (mm);

r_i — the cross-sectional raceway groove radius of inner ring (mm).

(3) Different factors for calculating basic static load rating for radial and thrust angular contact ball bearings

1) Angular contact radial ball bearings

In the calculation of C_{0r} , the conformities between the balls and the raceways are

$$r_i/D_w \leq 0.52 \text{ and } r_e/D_w \leq 0.53$$

2) Angular contact thrust ball bearing

In the calculation of C_{0a} , the conformities between the balls and the raceways are

$$r_i/D_w \leq 0.54 \text{ and } r_e/D_w \leq 0.54$$

(4) Comparison of adjusted basic static axial load ratings C_{0ar} and C_{0aa} for radial and thrust angular contact ball bearings

1) General

For certain applications, angular contact ball bearing with contact angle $\alpha \leq 45^\circ$ and $\alpha > 45^\circ$ are manufactured with the same conformities between the balls and the raceways, and sometimes there is a need to calculate and also to compare their true load ratings.

The basic static load ratings C_{0r} and C_{0a} can be calculated using International Standard or taken from a bearing catalog if available from that source.

However, as described above, C_{0r} and C_{0a} are calculated with different conformities for radial and thrust bearings. If a correct calculation and comparison is to be made, C_{0r} and C_{0a} have to be recalculated to the adjusted basic static axial load rating C_{0ar} and C_{0aa} based upon the same conformities.

The recalculation can be performed using the above equations of this annex for the two different conformities — radial bearing conformities and thrust bearing conformities respectively.

Comparison of load ratings is mainly of interest for bearing intended to operate in applications where axial loads are predominant, and therefore comparison of basic static axial load rating is dealt with in this annex.

The contact angle α is assumed to be constant, independent of the axial load, which means that the accuracy is reduced for bearings with small contact angles, subjected to heavy loads.

2) Angular contact ball bearings with radial bearing conformities

$$(r_i/D_w \leq 0.52 \text{ and } r_e/D_w \leq 0.53)$$

$$C_{0ar} = C_{0r}/Y_0$$

$$C_{0aa} = 1.43 C_{0a}$$

3) Angular contact ball bearings with thrust bearing conformities

$$(r_i/D_w \leq 0.54 \text{ and } r_e/D_w \leq 0.54)$$

$$C_{0ar} = 0.7 C_{0r}/Y_0$$

$$C_{0aa} = C_{0a}$$

(5) Examples:

1) Angular contact ball bearings with $\alpha = 45^\circ$

Compare the adjusted basic static axial load ratings of a single-row angular contact ball bearing with $\alpha = 45^\circ$, when it is regarded as a radial bearing and as a thrust bearing. For the selected bearing $(D_w \cos \alpha)/D_{pw} = 0.16$ and $i = 1$. The bearing has radial bearing conformities.

As a radial bearing:

C_{0r} is calculated according to the above equation, i. e. $C_{0r} = f_0 Z D_w \cos \alpha$.

According to Table 2-47, $f_0 = 14.9$ and according to Table 2-48, $Y_0 = 0.22$

$$C_{0r} = 14.9 Z D_w \cos 45^\circ = 10.54 Z D_w$$

By inserting C_{0r} and Y_0 into the above relating equation:

$$C_{0ar} = 0.7 C_{0r} / Y_0 = 10.54 Z D_w 0.22 = 47.9 Z D_w$$

As a thrust bearing:

C_{0a} is calculated according to the above equation, i. e. $C_{0r} = f_0 Z D_w \sin \alpha$.

According to Table 2-47, $f_0 = 48.8$.

$$C_{0aa} = 14.3 * 48.8 Z D_w \sin 45^\circ = 49.3 Z D_w.$$

These calculations show that the basic static load rating $C_{0ar} \approx C_{0aa}$, which confirms that there is no discontinuity.

2) Angular contact ball bearings with $\alpha = 40^\circ$

Calculate the adjusted basic static axial load rating C_{0ar} of a single-row angular contact ball bearing with the the contact angle $\alpha = 40^\circ$. The bearing has thrust conformities. $D_w/D_{pw} = 0.091$, ball diameter $D_w = 7.5\text{mm}$, number of rows of ball = 1 and the number of balls $Z = 27$.

According to Table 2-47, for $(D_w \cos 40^\circ) / D_{pw} = 0.091 \cos 40^\circ = 0.07$, and then $f_0 = 16.1$. From Table 2-48, $Y_0 = 0.26$.

The above equation gives;

$$C_{0r} = f_0 Z D_w \cos \alpha = 16.1 * 27 * 7.5 \cos 40^\circ = 18731.$$

Note: This load rating is based on radial bearing conformities.

According to the above equation:

$$C_{0ar} = 0.7 * 18730 / 0.26 = 50430$$

$$C_{0a} = 50400\text{N}$$

3) Angular contact ball bearings with $\alpha = 60^\circ$

Calculate the adjusted basic static axial load rating C_{0aa} of a single-row angular contact ball bearing with the the contact angle $\alpha = 60^\circ$. The bearing has thrust conformities. $D_w/D_{pw} = 0.091$, ball diameter $D_w = 7.5\text{mm}$, and the number of balls $Z = 27$.

According to Table 2-47, for $(D_w \cos 60^\circ) / D_{pw} = 0.091 * \cos 60^\circ =$

0.046, and then $f_0 = 57.82$.

According to above equation:

$$C_{0a} = f_0 Z D_w \sin \alpha = 57.82 * 27 * 7.5 * \sin 60^\circ = 70649$$

Note: This load rating is based on thrust bearing conformities.

According to the above equation:

$$C_{0aa} = C_{0a} = 76049$$

$$C_{0aa} = 76000\text{N}$$

Chaper 3 Rolling Bearing Application Design

Rolling bearings are industrial products of nearly overall standardization and large amounts of production. Generally, the users of rolling bearings only need to properly select the right bearings. There are many aspects for rolling bearing selection involving bearing type, bearing accuracy, bearing model and bearing clearance. Bearing type and accuracy selection is determined by working conditions and usage requirements, But concentering the bearing model selection, we need to carry the calculation according to the structure design, and at last, to carry the bearing life calculation for determination. For bearing clearance selection, we also need to estimate and select according to the standards.

Rolling bearings are the core components for a support, During the process of constructing a support, in order to make it function normal as a support, we need to resolve many problems in the rolling bearing application design. For example, which kind of fits will be selected on the matching surfaces of the bearing inner ring bore diameter with the shaft outer diameter (OD) and the bearing outer ring outside diameter (OD) with the housing inner diameter; how does the bearing inner ring locate and fix on the shaft and the bearing outer ring locate and fix in the housing bore; which kind of the manufacturing accuracy and roughnesses should be selected for the shaft part and the housing bore part that surfaces will be matching with the bearing inner ring bore surface and the bearing outer ring outside surface respectively; how to decide the mounting dimensions and seal of the matching portions for the bearing axial locations.

The spindle of a machine tool is a mechanical executing unit, which plays a very important role for the manufacturing accuracy of machining parts. Therefore, there are different requirements for the running accuracy, axial location accuracy and running performance of the spindle. Designing the support structure reasonably would have a very important effect on the

above technical characteristics. We should also consider how to adjust the bearing clearance if selecting the angular contact and thrust contact bearings.

During the support structure design process, the strength calculation and the structure design are carried out alternately. It is the prerequisite for the shaft strength calculation to decide the support positions relating to the acting points of the external forces. It also is the basis to decide the shaft diameters of the support satisfying shaft strength requirement for the bearing life calculation and selection of the bearing model. This chapter mainly introduces the problems related to bearing application design, mainly the rolling bearing support structure types, the rolling bearing matching fit, the manufacturing accuracy for the matching parts with rolling bearings, the bearing clearance selection, the rolling bearing seal devices, the rolling bearing axial tightening devices, the rolling bearing installation dimensions and the calculation examples of rolling bearing support structure.

3.1 Rolling Bearing Support Structure

The spindle and the transmission shaft are the main components of a machinery. Their running conditions not only influence the working performance of a machinery, but also influence the main technical features of a machinery. So, designing the support structure reasonably has a very important influence on ensuring shaft running accuracy, keeping normal running and totally exercising bearing performance.

3.1.1 Considerations about selecting support structure

1. It should be considered firstly how to limit the shaft positions, including the radial position and the axial position. Generally, the shaft is supported by a two support structure. The shaft radial position is located by the two supports altogether.

There should be a radial or an angular contact bearing playing the role of locating the radial position of the shaft on each support. The axial positions can be located by the two supports altogether, or can be located by one axial direction of one support and by the other axial direction of the

other support.

Usually, the different axial locating mode can get the different running accuracy. Therefore, when designing support structure, it should be evident to select the axial locating method according to shaft running accuracy and working condition.

2. On the condition of subjecting to both radial load and axial load at the same time, support always uses duplex pair mounting of angular contact bearings and tapered roller bearings. There are three arrangement modes for angular contact bearings in pairs as shown in Figure 3 - 1. The back to back pair mounting mode that the one bearing outer ring wide back face faces to the other bearing outer ring wide back face, is shown in Figure 3 - 1a. In this arrangement mode, the force acting points of the two supports are located in the outside range of the support span. This arrangement mode has been getting comprehensive usage because the support span is large, the rigidity is good when the shaft hangs up, and when the shaft expands owing to temperature rise, the bearing outer rings and inner rings are disengaged relative to each other.

Therefore, the shaft can avoid the stalck. But if adopting preload mounting in this arrangement mode, the preload amount will be reduced due to temperature rise. The face to face pair mounting that one bearing outer ring narrow front face faces to the other bearing outer ring narrow front face, is shown in Figure 3 - 1b. In this arrangement mode, the force acting points of the two supports are located within the internal range of the support span. This arrangement mode mainly for short shaft and low temperatwe rise has also been getting comprehensive usage because the support structure is simple and convenient to mount or to dismount and easy to adjust.

But in this arrangement mode, it should be taken into account that there must be some spare clearance. The axial clearance must not be too large, which could cause the shaft running accuracy reducing. When the axial load is so heavy that requires multiple bearings to carry out the load altogether, we always use the tandem arrangement, in that one bearing outer ring wide face faces to the other outer ring narrow face, as shown in

Figure 3 – 1c. When adopting this tandem arrangement, the force acting points of the supports are located at the same side of the support span, which is called as the same side arrangement or tandem arrangement. When adopting this tandem arrangement, it must take care to guarantee even loading for every bearing from the aspects of manufacturing and support structure.

3. Generally, under machinery operation running, the temperature of the spindles or the transmission shafts are higher than that of the neighbouring parts, therefore, the spindles and shafts will be expanded and becoming longer. In order to keep the flexibility of the shaft rotation, during the support design, besides satisfying the requirements for axial locating accuracy, we also should satisfy the requirements for the shaft free expanding and shrinking when the heat generated by friction causes internal temperature rise. The axial locating way should correspond to the axial expanding and shrinking way.

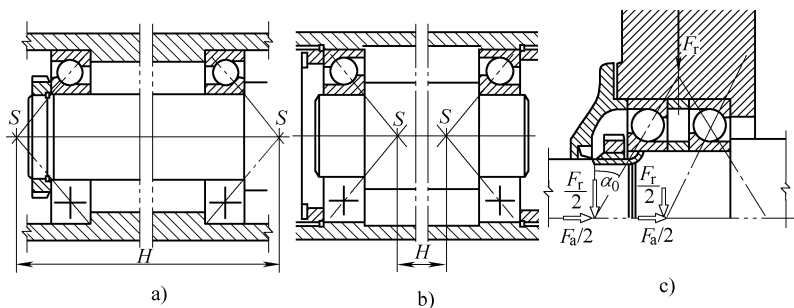


Figure 3 – 1 Three arrangement modes of angular contact bearings

a) back-back b) face-face c) tandem

4. In order to control the shaft running accuracy, it is necessary to adjust the bearing clearance. In order to meet the special requirements of engaged transmissions, it is necessary to adjust the axial location of the shaft. For example, in worm-gear drive, the axis of the worm must be located in the middle plane of the worm gear to guarantee correct engagement, therefore, the axis position of the worm gear can be adjusted in the axial direction; in the tapered gear transmissions, the two apexes of the

two tapered gear pitch circle cones must be coincided, therefore, the two tapered gear shafts can be adjusted in the axial direction.

3.1.2 Basic Support StructureTypes

Rolling bearing support structures can be divided as three basic types according to the capacity that the supports limit axial movement.

1. Supports limiting single direction movement

These support structures can only limit single direction axial movement and carry out single direction axial load. The different bearing arrangements and the different axial fixation structures are listed in Table 3 – 1.

2. Supports limiting double direction movement

These support structures can limit double direction axial movement and carry out double direction axial load, which are called as fixed supports, listed in Table 3 – 2. The bearing arrangement can be one bearing or two bearings. The support structures limiting double direction movement should make the bearings fix in double axial directions relative to the shaft and the housing bore. Most of the supports limiting double direction movement adopt the sleeve cup structures in order for convenient mounting and dismounting and axial adjustment.

3. Floating supports

These floating supports only play the role in the radial locating and carrying the radial load, as listed in Table 3 – 3. The floating supports can not make the axial locating and carry any axial load. When using the non-separable bearings with floating supports, the inner rings should be fixed on the shaft in double axial directions or the outer rings should be fixed in the housing in bores double axial directions, the bearing with the shaft altogether can be floating axially in the housing bore or the shaft can be floating axially relative to the bearing bore. When using cylindrical roller bearings, both the inner rings and outer rings should be fixed in double directions on the shafts and in the housing bores respectively, and it must take into account that the axial positions of the bearing inner ring and the outer ring can be adjusted for compensating the errors in manufacturing and mounting.

Table 3-1 Examples of the supports limiting single direction movement

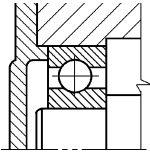
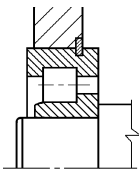
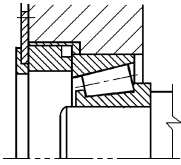
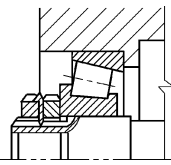
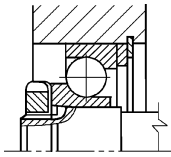
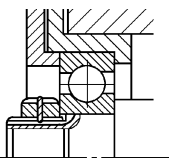
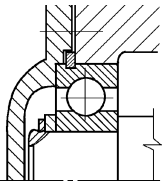
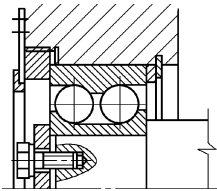
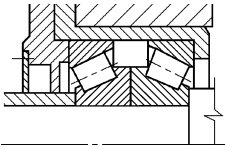
Order	Sketch	Description
1		The single direction axial load is transferred to the bearing inner ring and outer ring by the lateral abutment surface of the shaft, and at last it is carried by the end cap. The axial direction fixation of a radial ball bearing is achieved by the lateral abutment surface of the shaft and the end cap.
2		The single direction axial load is transferred by the lateral abutment surface of the shaft through the bearing inner ring rib to the outer ring, and at last it is carry by the snap ring. The axial direction fixation of the bearings is achieved by the lateral abutment surface of the shaft and the snap ring.
3		The single direction axial load is transferred through the lateral abutment surface of the shaft to the bearing inner ring and the outer ring, and at last it is carried by the screw ring. This tapered roller bearing axial fixation is achieved by the lateral abutment surface of the shaft and screw ring.
4		The single direction axial load is transferred through the double screw nuts on one shaft end to the bearing inner ring and the outer ring, and at last it is carried by the rib of housing bore. This tapered roller bearing axial fixation is achieved by the lateral rib surface of the housing bore and the lock nuts.
5		The single direction axial load is transferred through the locknut on one shaft end to the bearing inner ring and the outer ring, and at last it is carried by the elastic retaining ring. This angular contact ball bearing axial fixation is achieved by the locknut and the elastic retaining ring.

Table 3-2 Examples of the supports limiting double direction movement

Order	Sketch	Description
1		For radial ball bearing subjected to double direction axial loads, the inner ring should be fixed in double axial directions by the lateral abutment surface of the shaft and the double locknuts; the outer ring should be fixed in double axial directions by the lateral abutment surface of the sleeve cup and the end cap.
2		For radial ball bearing subjected to double direction axial loads, the inner ring should be fixed in double axial directions by the lateral abutment surface of the sleeve cup and the elastic remaining ring; the outer ring should be fixed in double axial directions by the lateral abutment surface of the snap ring and the sleeve cup end cap.
3		For double-row angular contact ball bearing subjected to double direction axial loads, the inner ring should be fixed in double axial directions by the lateral abutment surface of the shaft and the pressing plank; the outer ring should be fixed in double axial directions by the elastic remaining ring and the screw ring.
4		The fixing end of the support is composed of two tapered roller bearings in face-to-face arrangement. The two bearing inner rings should be fixed in double axial directions by the sleeve cup and the lateral abutment surface of the shaft; the two outer rings should be fixed in double axial directions by the lateral abutment surface of the sleeve cup and the end cap. For the face-to-face arrangement of angular contact bearing, the fix method is as same as the above.

(continued)

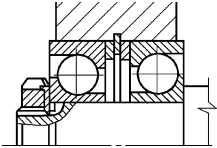
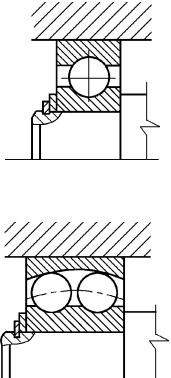
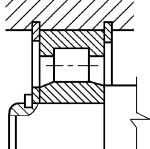
Order	Sketch	Description
5		The fixing end of the support is composed of two angular contact ball bearings in back-to-back arrangement. The two bearing inner rings should be fixed in double axial directions by the lateral abutment surface of the shaft and the locknut on the shaft end; the two outer rings should be fixed in double axial directions by an elastic retaining ring in a bore between the two outer rings. For face-to-face arrangement, the fix method is as same as the above.

Table 3 – 3 Examples of the floating supports

Order	Sketch	Description
1		Used in non-separable radial ball bearing or aligning radial ball bearings. In this case, the inner ring should be fixed in double axial directions. The outer ring can be floating in the bearing housing bore, therefore, the fit between the bearing outer ring and the bearing housing bore should select the loose fit.
2		Used in separable radial cylindrical roller bearing. In this case, both the inner ring and the outer ring need to be fixed in double axial directions. When the shaft is expanded due to heat, the shaft with the inner ring can be floating relative to the outer ring.

3. 1. 3 Three Basic Shaft Support Structure Modes

According to the shaft concrete structure and the usage requirement,

the two shaft supports can use two supports limiting single axial direction movement as shown in Figure 3-2. When the support is mainly subjected to the radial load, the support can adopt the deep groove ball bearing. In this case, the matching fit between the bearing outer ring and the housing bore should select the loose fit, and there should leave 0.5 ~ 1 mm clearance between the bearing end face and the end cap. When the support is subjected to the combined radial load and axial load, the supports mostly adopt the angular contact ball bearing or the tapered roller bearing. In order to compensate the shaft length expansion due to heat, there must be some axial clearance. For high speed spindles subjecting to light load, such as the grinding wheel shaft of grinding machine that adopts this type of support, in order to remove the floating clearance and improve the running accuracy, we always use the axial preload by elastic spring.

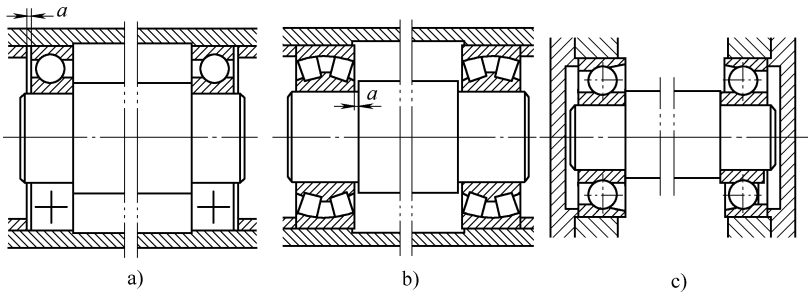


Figure 3-2 Support combinations limiting single axial direction position

Note: a is the spare clearance for compensating shaft length expansion due to heat.

Generally, for the applications of the spindles of precision machine tool or the applications that require high speed, high temperature and high accuracy and carry large load, it is mostly to select the combined support that one end support is fixed and the other end support is floating. For the combined support as shown in Figure 3-3, the fixing end can use one bearing that can limit the shaft positions in radial direction and in double axial directions, and also can use two identical bearings or two different bearings to limit the

shaft position in double axial directions. For floating support, there are two types: to use the non-separable deep groove ball bearing, aligning ball

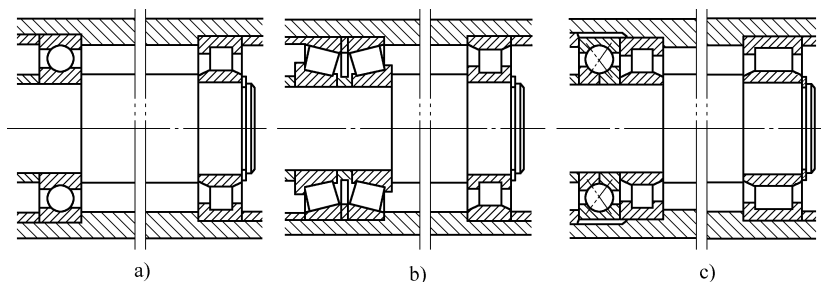


Figure 3-3 Fixing support—floating support combination

bearing or aligning roller bearing; or to use the separable cylindrical roller bearing. This combined support that one end is fixed and the other end is floating has been getting very comprehensive usage. In some special applications, if there is a requirement that the shaft axial position should be automatically adjusted to compensate the non-even wear in manufacturing and mounting, the combined floating support can be used in two shaft ends, that is not limiting the shaft position in axial directions. For example, in the Herringbone gear transmission, because the larger gear shaft has been fixed in the axial directions and the larger and smaller herring bone gears are engaged, the smaller gear can not be going out. The free floating of the smaller herring bone gear shaft could avoid the non-even wear on the gear tooth. Figure 3-4 are the two floating support combination.

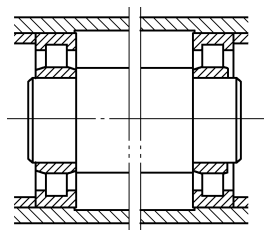
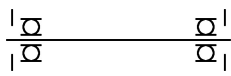
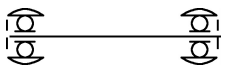
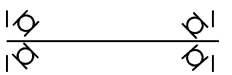
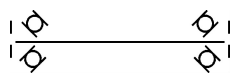
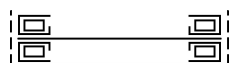
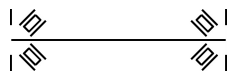
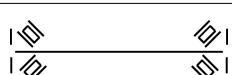


Figure 3-4 Floating support—floating support combination

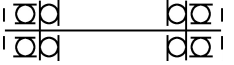
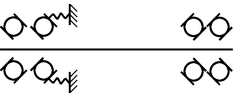
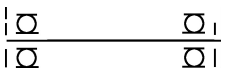
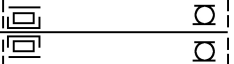
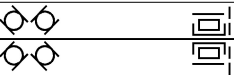
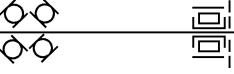
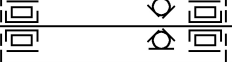
3.1.4 Common Support Structure Sketch

The common support structure sketches are listed in Table 3-4. The symbols used in Table 3-4 are illustrated in Table 3-5.

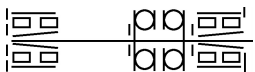
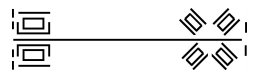
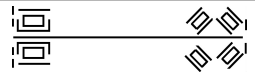
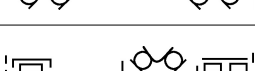
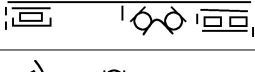
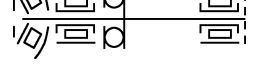
Table 3-4 Common support structure sketches

Support mode	Order	Bearing arrangement	Conditions subjecting to axial load	Compensation way for shaft expansion due to heat	Other features	
double end fixed support (suitable for small span, small temperature difference. For ball bearing, $l/d < 3 \sim 8$; for roller bearing, $l/d < 4 \sim 12$; for small diameter, we can take larger value)	1		subjecting to single direction axial load (pointing to no clearance end)	the clearance between the outer ring end face and end cap	high speed, simple structure, easy adjustment	
	2		subjecting to double direction axial load	bearing floating clearance		
	3					
	4					
	5		subjecting to light double direction axial load	the clearance between the outer ring end face and end cap	simple structure, easy adjustment	
	6			bearing floating clearance		
	7					

(continued)

Support mode	Order	Bearing arrangement	Conditions subjecting to axial load	Compensation way for shaft expansion due to heat	Other features
double end fixed supports (suitable for small span, small temperature difference. For ball bearing, $l/d < 3 \sim 8$; for roller bearing, $l/d < 4 \sim 12$; for small diameter, we can take larger value.	8		subjecting to double direction axial load	bearing floating clearance	suitable for low speed vertical shaft
	9			bearing floating clearance increases after shaft expansion; preload amount kept by preload spring.	suitable for higher speed
	10			bearing floating clearance	improve rigidity of supports by radial preload
one end fixed support, the other support end floating (suitable for larger span and larger-temperature difference)	11			clearance fit between the right end radial ball bearing outer ring and bearing housing bore	high speed, simple structure, easy adjustment
	12			relative axial movement between the roller and outer ring raceway	simple strycture, easy adjustment
	13				improve rigidity of supports by axial preload
	14				
	15			axial movement of the left support roller relative to the outer ring raceway	higher speed, larger radial load capacity, structure compact

(continued)

Support mode	Order	Bearing arrangement	Conditions subjecting to axial load	Compensation way for shaft expansion due to heat	Other features
one end fixed support, the other support end floating (suitable for larger span and larger temperature differ- ence)	16		subjecting to axial load in two directions	axial movement of the left support's roller relative to the outer ring raceway	can subject to heavier radial and axial loads, good rigidity of supports
	17				can subject to heavier radial and axial loads, simple structure, easy ad- justment
	18				
	19			slide fit in the right support's bearing outer ring and housing bore	higher speed
	20			axial movement be- tween the left roller and inner ring raceway	high running accuracy, high capaci- ty of radial and axial load, good rigidi- ty
	21		subjecting to axial load in two directions, one direction axial load can be heavier	axial movement of the right roller relative to outer ring raceway	suitable for cantilever subjecting to axial load in two directions
	22		subjecting to lighter axial load in two direc- tions	axial movement of the right rollers relative to the outer ring raceway	suitable for mainly subjecting to radial load at multiple support shaft, simple structure, easy assembly and disassembly

(continued)

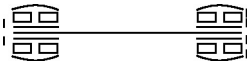
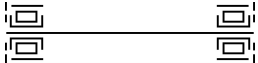
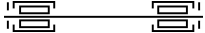
Support mode	Order	Bearing arrangement	Conditions subjecting to axial load	Compensation way for shaft expansion due to heat	Other features
one end fixed support, the other support end floating (suitable for larger span and larger heat expansion)	23		subjecting to lighter axial load in two directions	clearance fit between the right outer ring and housing bore	suitable for the shaft subjecting to heavier load, good center alignment
supports with two end floating	24		can not subject to axial load	movement between the two end rollers relative to the outer ring raceways	suitable for shaft floating in axial direction
	25			axial movement between the two end needle rollers relative to the shaft	

Table 3 – 5 Symbol illustration

Symbol	Illustration	Symbol	Illustration
	deep groove ball bearing		single-direction thrust ball bearing
	insert bearing		double-direction thrust ball bearing
	angular contact ball bearing		four-point contact ball bearing
	aligning ball bearing		double-direction thrust angular contact ball bearing
	cylindrical roller bearing (outer ring without rib)		needle roller bearing without inner ring
	cylindrical roller bearing (outer ring with rib)		tapered roller bearing
	cylindrical roller bearing (inner ring without rib)		thrust aligning roller bearing
	double-row cylindrical roller bearing (tapered bore)		preload spring
	aligning roller bearing		should be the fixed bearing ring in axial directions
			shaft

3. 2 Rolling Bearing Matching Fit Selection

The shaft running accuracy is related to the bearing running accuracy and the support locating accuracy. When adopting radial bearing or angular

contact bearing as support (except to non-radial locating), there should be interference fit in bearing inner ring with shaft and bearing outer ring with housing bore, which not only can guarantee the radial locating accuracy of the support, but also can prevent the sliding and wear between bearing inner ring and shaft, and outer ring and housing bore; at the same time, which also can guarantee overall even contacting of inner ring and outer ring with supporting surfaces. The interference amount of matching surfaces varies after mounting and in operation, and the more the deformation is due to load and temperature difference, the more the interference fit amount varies. Under general operation condition, the inner ring becomes larger because of the tangential deformation and the heat expansion at the loading surface of the inner ring raceway, Therefore, the matching fit of the inner ring with shaft becomes loose; the interference fit amount in the inner ring with shaft becomes smaller. The interference fit amount becomes larger because of the outer ring expansion, so the matching fit becomes tight. Therefore, while selecting matching fit, usually selecting the tighter matching fit for inner ring with shaft, and selecting the looser matching fit for outer ring with housing bore. So long as keeping the effective interference fit amount not smaller than zero, then the support locating accuracy requirement can be met and can guarantee the bearing load carrying capacity fully developed. However, the effective interference fit amount also can not be too large. Because both the bearing inner ring and outer ring are thin wall precision components easily developing elastic deformation, under the larger interference fit amount and load, the surface shapes of bearing inner ring and outer ring race ways will approach to the support surface shapes easily, therefore, this condition can influence bearing running accuracy and cause vibration and noise. The larger interference fit amount also can cause the bearing working clearance reduced and bearing mounting and dismounting in convenient. For adopting non-separable bearing as floating support, clearance fit on the floating position of bearing rings should be selected. For the bearings in heavy machine, the interference fit should not be selected.

The rolling bearing fit standard defines the recommended practice of tolerance band positions and ranges in the fit between bearing inner rings and shafts and outer rings and housing bores. The fit symbols consist of English letter and number expressing the fitting feature and the tolerance grade of fitting component respectively. In order to meet the requirements of support accuracy, bearing utilizing performance and life, the fit standard also defines the geometric tolerances and surface roughness for the matching position of the matching components. When selecting fit, it is the right way to select the proper fit according to the recommended standard, and it also is the right way to estimate the minimum interference fit amount, and then to select the proper fit according to the minimum interference fit amount. When interference fit largely influences the bearing floating clearance, we should estimate the influencing range for the bearing floating clearance. Here also lists the bearing tolerances and the tolerances of matching shafts and housing bores for using in estimate.

3.2.1 Rolling Bearing Fit Characteristics

The rolling bearing standard is an industrial special standard, which characteristics are as follows:

1. The bearing inner ring bore diameter d , outer ring outer diameter D are the bearing nominal dimensions for the fits between the bearing and shaft, and bearing and the housing bore. The bearing inner ring bore diameter d takes the hole basis system, which tolerance band is located in the lower part of zero line, that is its upper deviation is zero, and its lower deviation is negative. Generally, comparing with other hole basis system that the tolerance band is located on the upper part of the zero line, under the same fitting, it is easy to get more tightened fit. The bearing outer diameter D takes the shaft basis system, which tolerance band is located in the lower part of zero line that is as same as the hole basis system, that is its upper deviation is zero, and its lower deviation is negative. Generally, the fit between bearing and hole is looser, but comparing with the same name fit of other shaft basis system, the tolerance band is different. The varied precision grades of bearing tolerance band are shown in Figure 3-5.

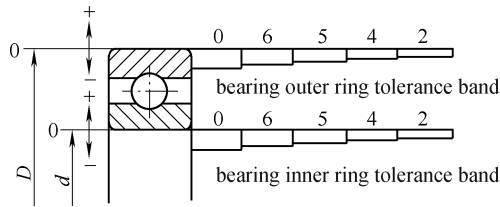


Figure 3-5 Bearing inner ring and outer ring tolerance band

2. Generally, the manufacturing accuracy of matching components should correspond to the bearing accuracy. Usually, the shaft manufacturing accuracy should take the same grade accuracy as the bearing accuracy or one higher grade accuracy than the bearing accuracy; But the housing bore manufacturing accuracy should take one lower grade accuracy than the bearing accuracy or the same grade accuracy as the bearing accuracy. For different grade accuracy bearings, the fitting standard defines the recommended corresponding geometric tolerances and the surface roughness of the matching fit between the shaft and housing bore. The tolerance grade of the shaft matching with the grade 0 or grade 6 accuracy bearing usually IT6; the housing bore is IT7.

For the applications having high level requirements for running accuracy and smooth rotation, the bearing accuracy grade should be increase, at the same time, the correspondent accuracy grade of the matching component also need to increase.

3. The fit of the hollow shaft and the housing made by light metal or cast iron is more tighter than the fit of the solid shaft and the housing made by cast steel. Therefore, usually we first select the fit for the bearing and housing bore according to the solid shaft and the housing made by cast steel, and then according to the introduction method in Section 3.2.5 of this chapter to calculate a required tighter fit.

3.2.2 Commonly Used Tolerance Band of Fits for Rolling Bearing with Shaft and Housing

1) The commonly used fits of varied accuracy grade bearings are listed in Table 3-6.

Table 3-6 The commonly used fits of varied accuracy grade bearings

Accuracy	Bearing with shaft		Bearing with housing bore		
grade	transition fit	interference fit	clearance fit	transition fit	interference fit
grade 0	h9				
	h8	r7、 k6、	H8	J7、 JS7、 K7、 M7、 N7	P7
	g6 h6 j6 js6	m6、 n6、 p6、	G7、 H7	J6、 JS6、 K6、 M6、 N6	P6
	g5 h5 js5	r6、 k5、 m5	H6		
grade 6	g6 h6 j6 js6	r7、 k6、 m6、	H8	J7、 JS7、 K7、 M7、 N7	P7
	g5 h5 js5	n6、 p6、 r6、 k5、 m5	G7、 H7 H6	J6、 JS6、 K6、 M6、 N6	P6
grade 5	h5 js5	k6、 m6	G6、 H6	JS6、 k6、 M6	
		k5、 m5		JS5、 k5、 M5	
grade 4	h5 js5	k5 m5	H5	K6	
	h4 js4	k4		JS5、 K5、 M5	
grade 2	h3 js3		H4	JS4、 K4	

Note: The fits of bore N6 with grade 0 accuracy bearings (outer diameter $D < 150\text{mm}$) and grade 6 accuracy bearings are transition fit.

Shaft r5 is used for bore diameter $d > 120 \sim 500\text{mm}$; Shaft r7 is used for bore diameter $d > 180 \sim 500\text{mm}$.

2) Figure 3 – 6 and Figure 3 – 7 are the fit examples of commonly used tolerance band distribution of grade 0 and grade 6 bearings with shafts and housing bores.

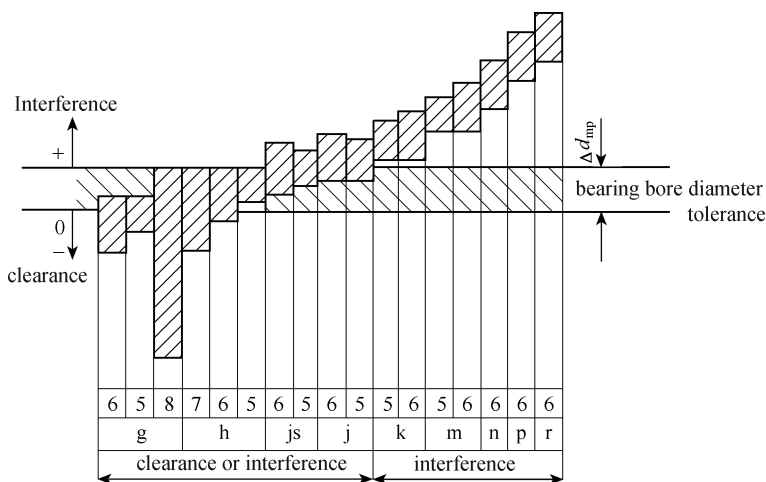


Figure 3-6 Bearing with shaft commonly used fit tolerance band distribution

Note: Δd_{md} is the average bore diameter deviation of the bearing inner ring single plane.

(3) Bearing dimension Along with the increase of bearing dimension, the interference fit amount increases; the clearance fit amount also increases.

(4) Bearing floating clearance The bearing floating clearance will be decreased due to interference fit. Therefore, in order to select the proper fit and the bearing floating clearance, after installation, the bearing floating clearance should be inspected to see if the bearing floating clearance could meet the requirements of the application.

(5) Other factor effects Many factors can influence the fit selection, such as the materials, strengths and thermal conductivity of the shaft and the housing, the outside surroundings, the heat generated in bearing and heat transfer, the installation and adjustment features of the support.

3.2.4 Recommended Fits by Rolling Bearing Fit Standard

1) Table 3 – 8 recommends the tolerance bands of shaft, and fits of radial bearing inner ring and shaft.

2) Table 3 – 9 recommends the tolerance bands of hole, and fits of radial bearing and housing bore.

3) Table 3 – 10 recommends the tolerance bands of shaft, and fits of thrust bearing shaft.

4) Table 3 – 11 recommends the tolerance bands of hole, and fits of thrust bearing and housing bore.

5) Table 3 – 12 recommends the 2 grade accuracy bearing fits.

Table 3 – 8 Radial bearing and shaft fit, shaft tolerance band symbols

Cylindrical bore bearing					
Running condition	Load condition	Deep groove ball bearing, self-aligning ball bearing, angular contact ball bearing	Cylindrical roller bearing, tapered roller bearing	Self-aligning	Tolerance band

(continued)

illustrition	example		bearing nominal bore diameter/mm				
rotating inner ring load and oscillating load	common ma- chinery, motor, spindle, pump, internal-com- busion engine, gear transmis- sion, railway axle box, crusher	light load	≤18	—	—	h5	
			> 18 ~ 100	≤40	≤40	j6 ^①	
			> 100 ~ 200	> 40 ~ 140	> 40 ~ 100	k6 ^①	
			—	> 140 ~ 200	> 100 ~ 200	m6 ^①	
		normal load	≤18	—	—	j5	js5
			> 18 ~ 100	≤40	≤40	k5 ^②	
			> 100 ~ 140	> 40 ~ 100	> 40 ~ 65	m5 ^②	
			> 140 ~ 200	> 100 ~ 140	> 65 ~ 100	m6	
			> 200 ~ 280	> 140 ~ 200	> 100 ~ 140	n6	
			—	> 200 ~ 400	> 140 ~ 280	p6	
			—	—	> 280 ~ 500	r6	
			heavy load		> 50 ~ 140	> 50 ~ 100	n6 ^③
		> 140 ~ 200		> 100 ~ 140	p6		
		> 200		> 140 ~ 200	r6		
		—		> 200	r7		
fixed inner ring load	stationary shaft wheel, tension, wheel, vibrator	all loads	all dimensions			f6 ^①	
						g6	
						h6	
						j6	
only axial load			all dimensions			j6、js6	
tapered bore bearing							
all loads	railway axle box		all dimensions mounting in withdrawal sleeve			h8 (IT6) ^{④、⑤}	
	mechanical transmission		all dimensions mounting in adapter sleeve			h9 (IT7) ^{④、⑤}	

- Note: ① For high accuracy applications, we should use j5, k5 instead of j6, k6... .
- ② The fits of tapered roller bearing and angular cotact ball bearing have only little influence on floating clearance. we can use k6, m6 instead of k5, m5.
- ③ Under heavy load, bearing floating clearance should select the group clearance bigger than group 0 clearance.
- ④ For higher accuracy requirement or speed requirement, we should select h7 (IT5) instead of h8 (IT6).
- ⑤ IT6, IT7 designates the cylindricity tolerance.

Table 3 – 9 Radial bearing and housing bore fit, hole tolerance band symbols

running condition		load condition	other conditions	tolerance band ^①	
illustrition	example			ball bearing	roller bearing
fixed outer ring load	general machinery, railway axle box, motor, pump, main bearing of crankshaft	light, normal heavy	easy axial movement, split housing	H7、G7 ^②	
oscillating load		shocking	axial movement, entire or split housing	J7、JS7	
		light, normal			
		normal, heavy	no axial movement entire housing	K7	
		shocking		M7	
rotaing outer ring load	tension wheel, wheel hut bearing	light		J7	K7
		normal		K7、M7	M7、N7
		heavy		—	N7、P7

Note: ① For co-listing tolerance band, along with the dimension increasing, we should select from left to right, For higher requirement of running accuracy, we can raise the tolerance band grade one level higher than that of normal.

② Not suitable for split housing.

Table 3 – 10 Thrust bearing and shaft fit, shaft tolerance band symbols

Running condition	Load condition	Thrust ball and thrust roller bearing	Self-aligning roller bearing ^②	Tolerance band
		nominal bearing bore diameter/mm		
only axial load		all dimensions		j6、js6
fixed shaft washer load	radial and axial combined load	—	≤250	j6
		—	>250	js6
rotating shaft washer load or oscillation load	axial combined load	—	≤200	k6 ^①
		—	>200 ~400	m6
		—	>400	n6

Note: ① For small interference amount, we can use j6, k6, m6 instead of k6, m6, n6 respectively.

② Including thrust tapered roller bearing, and thrust angular contact ball bearing.

Table 3 – 11 Thrust bearing and housing bore fit, hole tolerance band symbols

running condition	load condition	bearing type	tolerance band	note
only axial load		thrust ball bearing,	H8	
		thrust cylindrical, tapered roller bearing	H7	
		thrust self-aligning roller bearing		The clearance of housing bore and housing washer is $0.001D$ (D is bearing nominal diameter)
fixed housing washer load	radial and axial combined load	thrust angular contact ball bearing, thrust self-aligning roller bearing, thrust tapered roller bearing	H7	
rotating housing washer load or oscillating load			K7	common condition
			M7	larger radial load

Table 3 – 12 2 grade accuracy bearing fit

ring load type	fit	ring load type	fit
	inner ring to shaft		outer ring to housing bore
inner ring no rotating local load	js3、h3	outer ring no rotating local load	JS3、JS4、H3 H4
inner ring no rotating circulate load oscillating load	js3 js3	outer ring rotating circulate load oscillating load	K4 JS3、JS4

3.2.5 Fit Selection of Hollow Shaft and Light Metal Housing

When the wall thickness of hollow shaft is thin to a certain degree, that is the diameter ratio $C_i \geq 0.5$, the wall thickness of the hollow shaft just has some effect on the selection of fit. The effect degree is relative to the bearing inner ring diameter ratio C_e .

$$C_i = d_i/d$$

$$C_e = d/d_e$$

Here d_i —hollow shaft inner diameter (mm);
 d —bearing bore diameter (mm);
 d_e —bearing inner ring outer diameter (mm).

If the bearing inner ring outer diameter is unknown, we can calculate it by the following equation:

$$C_e = \frac{d}{K(D - d) + d}$$

Here d —bearing bore diameter (mm);
 D —bearing outer diameter (mm);
 K —bearing type constant.

$K = 0.25$ suitable for cylindrical roller bearing and self-aligning roller bearing;

$K = 0.30$ suitable for other bearings.

Figure 3-8 is the relationship of the interference amount of solid shaft and hollow shaft. After selecting the interference amount of solid shaft according to solid shaft and checking out the corresponding average interference amount ΔV , according to the bearing inner ring diameter ratio C_e , we can check out the corresponding $\Delta H/\Delta V$ value from the C_i in Figure 3-8, from ΔV we calculate the required average interference amount ΔH and then we check out the corresponding fit from ΔH .

[**Example 3-1**] The deep groove ball bearing 6208 mounted on the hollow shaft of diameter ratio $C_i = 0.8$, normal load condition, general accuracy, try to select the hollow shaft fit.

[**Resolve**] According to Table 3-8, select the k5 fit. From Table 3-25, find when the shaft fit k5 is of diameter as 40mm, the average interference amount ΔV equals to $(2 + 25) \mu\text{m}/2 = 13.5 \mu\text{m}$.

From $C_i = 0.8$ and

$$C_e = \frac{d}{K(D - d) + d} = \frac{40}{0.3(80 - 40) + 40} = 0.77$$

On Figure 3-8 curves found $\Delta H/\Delta V = 1.7$

$$\Delta H = 1.7 \Delta V = 1.7 * 13.5 \mu\text{m} = 23 \mu\text{m}$$

From Table 3-25 we can find that the fit is m6 when the average interference amount ΔV is $23 \mu\text{m}$. Therefore, we should select the m6 as the fit of the bearing inner ring to the hollow shaft.

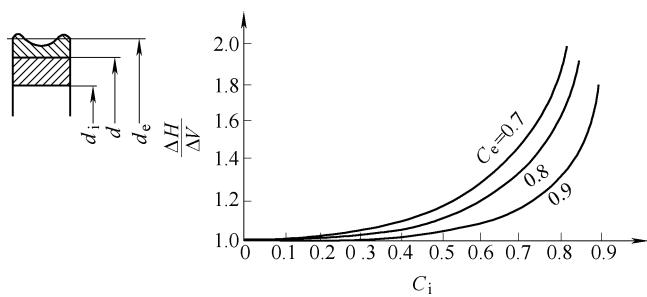


Figure 3-8 Relationship of interference amount for hollow shaft and solid shaft

3.2.6 Selecting Rolling Bearing Fit by Estimation

Generally, we select bearing fit according to the experience and conditions of stallation, adjustment and usage of the bearings on the machinery, by analogy, and referring to the standard’s recommendation. But we can also estimate the interference amount according to the equations listed in Table 3-13, using the required minimum interference amount as the basis for selecting fit or checking and calculating for the having selected minimum interference amount.

Table 3-13 Nominal interference amount Δd estimation

Name	Equation	Symbol
effective interference amount $\Delta d_y/\text{mm}$	(grinding) $\Delta d_y = \frac{d}{d+3}\Delta d$ (polishing) $\Delta d_y = \frac{d}{d+2}\Delta d$	Δd —nominal interference amount (μm) d —bearing nominal bore diameter (mm)
interference amount $\Delta d_F/\text{mm}$ reduced from F_r	$\Delta d_F = 0.25 \sqrt{\frac{d}{9.8B}} F_r$	B —bearing inner ring width (mm) F_r —radial load (N)
interference amount $\Delta d_T/\text{mm}$ reduced from ΔT	$\Delta d_T = (0.10 \sim 0.15) \Delta T * \alpha d * 1000$ $\approx 0.00015 \Delta T d$	ΔT —bearing interior and environment temperature difference ($^{\circ}\text{C}$)
steel thermal expansion coefficient $\alpha / (1/^{\circ}\text{C})$	$\alpha = 12.5 * 10^{-6} 1/^{\circ}\text{C}$	

(continued)

Name	Equation	Symbol
interference condition	$\Delta d_y - \Delta d_f - \Delta d_T \geq 0$	
nominal interference amount $\Delta d/\text{mm}$	$\Delta d \geq \frac{d+3}{d} \left(0.25 \sqrt{\frac{d}{9.8B}} F_r + 0.0015 \Delta T d \right)$	

Note: Generally, the temperature difference of inner ring and outer ring is about $5 \sim 10^\circ\text{C}$ when the bearings and their environment is not subjecting to hot or cold;

But for the bearings on wheel, the temperature difference of inner ring and outer ring is about $15 \sim 20^\circ\text{C}$ because they are subjecting to the airflow cooling when moving.

3.2.7 Bearing Fit Selection of Machine Tool Spindle

Normal accuracy (0 grade) bearings can meet the requirements of general machinery on support accuracy, such as gear box of common machine tools, automobiles and tractors and common motor, pump, compressor of ordinary machines that there are low requirements for running accuracy and under medium load and speed. But for precision machine tools, inspecting instruments, and high speed machinery that having high requirements for working precision and running smoothness, we have to use high accuracy bearings. Table 3-14 lists the bearing accuracy grades of the spindle of commonly used precision machine tool for reference.

Table 3-14 Bearing accuracy grades of machine tool spindle

Bearing type	Precision grade	Application
deep groove ball bearing	grade 4, grade 2	high precision grinding machine, tap grinding machine, thread grinding machine, gear grinding machine, insert gear knife grinding machine (2 grade)
angular contact ball bearing	grade 5	precision boring machine, bore grinding machine, gear machine tool
	grade 6	lying lathe, milling machine
double-row cylindrical roller bearing	grade 4	precision leading screw lathe, high precision lathe, high precision outside round grinding machine
	grade 5	precision lathe, precision milling machine, turret lathe, ordinary outside round grinding machine, more-shaft lathe, boring machine
	grade 6	lying lathe, automatic lathe, milling machine, vertical lathe

(continued)

Bearing type	Precision grade	Application
cylindrical roller bearing self-aligning roller bearing	grade 6	precise lathe and precision milling machine's rear bearing
tapered roller bearing	grade 2 , 4	coordinate boring machine (grade 2) , gear grinding machine (grade 4)
	grade 5	precision lathe, precision milling machine, boring machine, precision turret lathe, gear-hobbing machine
	grade 6X	milling machine, lathe
thrust ball bearing	grade 6	general precision lathe

The recommended fits for deep groove ball bearing, angular contact ball bearing and tapered roller bearing of grade 2, grade 4 accuracy are listed in Table 3 – 15 and Table 3 – 16.

Table 3 – 15 Recommended fits for deep groove ball bearing, angular contact ball bearing and tapered roller bearing of grade 4, grade 2 accuracy fitting to shaft

Nominal bearing bore diameter d/mm		grade 4	grade 2
over	to	shaft diameter tolerance band	
18	120	js4、h4 (preload)	js3、h3 (preload)
120	250	k4	

When using grade 2 accuracy bearing, it is the best method to manufacture the spindle shaft diameter and the housing bore according to the fit feature of the actual bearing bore diameter and the bearing outer diameter. It should be also ensured that the interference amount and the clearance amount are listed in Tables 3 – 17 and 3 – 18.

Table 3 – 16 Recommended fits for deep groove ball bearing, angular contact ball bearing and tapered roller bearing of grade 4, grade 2 accuracy fitting to housing bore

Bearing type	whether or not floating	grade 4	grade 2
		housing bore tolerance band	
deep groove ball bearing angular contact ball bearing	floating	H5	H4
	fixing	JS5	JS4

(continued)

Bearing type	whether or not floating	4 grade	2 grade
		housing bore tolerance band	
tapered roller bearing	floating	H5	H4
	fixing	K5	K4
double-row cylindrical roller bearing with tapered bore		K5	K4

Table 3-17 Interference amount for deep groove ball bearing, angular contact ball bearing and tapered roller bearing of grade 2 accuracy fitting to shaft (unit: μm)

Bearing speed factor $dn \leq$	Nominal bearing bore diameter/mm							
	18 ~ 30		30 ~ 50		50 ~ 80		80 ~ 120	
	mini.	max.	mini.	max.	mini.	max.	mini.	max.
200000	0	+3	0	+4	+1	+5	+1	+6
400000	0	+2	0	+2	0	+2	0	+3
800000	-1	+1	-1	+1	-1	+1	—	—

Note: “-” is clearance amount, “+” is interference amount.

Table 3-18 Fitting clearance amount for deep groove ball bearing, angular contact ball bearing and tapered roller bearing of grade 2 accuracy fitting to housing (unit: μm)

Bearing speed factor $dn \leq$	Nominal bearing bore diameter/mm							
	30 ~ 50		50 ~ 80		80 ~ 120		120 ~ 180	
	mini.	max.	mini.	max.	mini.	max.	mini.	max.

deep groove ball bearing and angular contact ball bearing

100000	-4	+4	-5	+5	-6	+6	-7	+7
200000	-1	+5	-1	+6	-1	+7	-1	+9
400000	0	+4	0	+5	0	+6	0	+7
800000	+1	+3	+1	+3	+1	+4	—	—

double-row cylindrical roller bearing and tapered roller bearing

100000	-6	+2	-7	+2	-8	+3	-9	+3
200000	-4	+4	-4.5	+4.5	-5	+5	-6	+6
300000	-2	+6	-2	+7	-3	+8	-3	+9

Note: 1. “-” is clearance amount, “+” is interference amount.

2. For high requirements of support rigidity and anti-vibration, we can select the fit

tighter than the above clearance range.

3. For special high speed bearing, in order to reduce the vibration generated from disequilibrium of spindle parts we also can select tighter fit.

3.2.8 Bearing Tolerance and Its Matching Shaft and Housing Bore Tolerance

Every grade accuracy of bearing tolerances are listed in Table 3 – 19 to Table 3 – 24; the matching shaft and housing bore tolerances and possibly existing interference amounts and clearance amounts are listed in Table 3 – 25 to Table 3 – 36.

Table 3 – 19 Radial bearing (except tapered roller bearing)
bore diameter tolerance Δ_{dmp} (unit : μm)

Nominal bearing bore diameter d/mm		grade 0	grade 6	grade 5	grade 4	grade 2
		Δ_{dmp}				
over	to	low	low	low	low	low
—	0.6	– 8	– 7	– 5	– 4	– 2.5
0.6	2.5	– 8	– 7	– 5	– 4	– 2.5
2.5	10	– 8	– 7	– 5	– 4	– 2.5
10	18	– 8	– 7	– 5	– 4	– 2.5
18	30	– 10	– 8	– 6	– 5	– 2.5
30	50	– 12	– 10	– 8	– 6	– 2.5
50	80	– 15	– 12	– 9	– 7	– 4
80	120	– 20	– 15	– 10	– 8	– 5
120	150					– 7
150	180					– 7
120	180	– 25	– 18	– 13	– 10	
180	250	– 30	– 22	– 15	– 12	– 8
250	315	– 35	– 25	– 18		
315	400	– 40	– 30	– 23		
400	500	– 45	– 35			
500	630	– 50	– 40			
630	800	– 75				
800	1000	– 100				

(continued)

Nominal bearing bore diameter d/mm		grade 0	grade 6	grade 5	grade 4	grade 2
		Δ_{dmp}				
over	to	low	low	low	low	low
1000	1250	-125				
1250	1600	-160				
1600	2000	-200				

Note: 1. The upper deviation of Δ_{dmp} is 0; from GB 307.1—2005.

2. Only the grade 2 bearings have dimension segments of 120 ~ 150 and 150 ~ 180, all others as 120 ~ 180.

Table 3-20 Radial bearing (except tapered roller bearing)
outer diameter tolerance Δ_{Dmp} (unit: μm)

Nominal bearing bore diameter D/mm		grade 0	grade 6	grade 5	grade 4	grade 2
		Δ_{Dmp}				
over	to	low	low	low	low	low
—	2.5	-8	-7	-5	-4	-2.5
2.5	6	-8	-7	-5	-4	-2.5
6	18	-8	-7	-5	-4	-2.5
18	30	-9	-8	-6	-5	-4
30	50	-11	-9	-7	-6	-4
50	80	-13	-11	-9	-7	-4
80	120	-15	-13	-10	-8	-5
120	150	-18	-15	-11	-9	-5
150	180	-25	-18	-13	-10	-7
180	250	-30	-20	-15	-11	-8
250	315	-35	-25	-18	-13	-8
315	400	-40	-28	-20	-15	-10
400	500	-45	-33	-23		
500	630	-50	-38	-28		
630	800	-75	-45	-35		
800	1000	-100	-60			
1000	1250	-125				

(continued)

Nominal bearing bore diameter D/mm		grade 0	grade 6	grade 5	grade 4	grade 2
		Δ_{Dmp}				
over	to	low	low	low	low	low
1250	1600	− 160				
1600	2000	− 200				
2000	2500	− 250				

Note: 1. The upper deviation of Δ_{Dmp} is 0 ;
2. from GB/T 307. 1—2005.

Table 3 – 21 Tapered roller bearing bore diameter tolerance Δ_{dmp}

(unit: μm)

Nominal bearing bore diameter d/mm		grade 0	grade 6X	grade 5	grade 4	grade 2
		Δ_{dmp}				
over	to	low	low	low	low	low
—	10	− 12	− 12	− 7	− 5	− 4
10	18	− 12	− 12	− 7	− 5	− 4
18	30	− 12	− 12	− 8	− 6	− 4
30	50	− 12	− 12	− 10	− 8	− 5
50	80	− 15	− 15	− 12	− 9	− 5
80	120	− 20	− 20	− 15	− 10	− 6
120	180	− 25	− 25	− 18	− 13	− 7
180	250	− 30	− 30	− 22	− 15	− 8
250	315	− 35	− 35	− 25	− 18	− 8
315	400	− 40	− 40	− 30		
400	500	− 45	− 45	− 35		
500	630	− 60	− 60	− 40		
630	800	− 75	− 75	− 50		
800	1000	− 100	− 100	− 60		
1000	1250	− 125	− 125	− 75		
1250	1600	− 160	− 160	− 90		
1600	2000					

Note: 1. The upper deviation of Δ_{dmp} is 0 ;
2. from GB/T 307. 1—2005.

Table 3 – 22 Tapered roller bearing outer diameter tolerance Δ_{Dmp}

(unit: μm)

Nominal bearing bore diameter D/mm		grade 0	grade 6X	grade 5	grade 4	grade 2
		Δ_{Dmp}				
over	to	low	low	low	low	low
—	18	– 12	– 12	– 8	– 6	– 5
18	30	– 12	– 12	– 8	– 6	– 5
30	50	– 14	– 14	– 9	– 7	– 5
50	80	– 16	– 16	– 11	– 9	– 6
80	120	– 18	– 18	– 13	– 10	– 6
120	150	– 20	– 20	– 15	– 11	– 7
150	180	– 25	– 25	– 18	– 13	– 7
180	250	– 30	– 30	– 20	– 15	– 8
250	315	– 35	– 35	– 25	– 18	– 9
315	400	– 40	– 40	– 28	– 20	– 10
400	500	– 45	– 45	– 33		
500	630	– 50	– 50	– 38		
630	800	– 15	– 75	– 45		
800	1000	– 100	– 100	– 60		
1000	1250	– 125	– 125	– 80		
1250	1600	– 160	– 160	– 100		
1600	2000	– 200	– 200	– 125		
2000	2500	– 250	– 250			

Note: 1. The upper deviation of Δ_{Dmp} is 0;

2. from GB/T 307.1—2005.

Table 3 – 23 Thrust bearing shaft washer diameter tolerance Δ_{dmp}

(unit: μm)

Nominal shaft washer bore diameter/mm		grade 0, grade 6, grade 5,		grade 4	
d (single direction) d_2 (double direction)		$\Delta_{\text{dmp}}, \Delta_{\text{d2mp}}$			
over	to	upper	low	upper	low
	18	0	– 8	0	– 7
18	30	0	– 10	0	– 8
30	50	0	– 12	0	– 10

(continued)

Nominal shaft washer bore diameter/mm		grade 0, grade 6, grade 5,		grade 4	
d (single direction) d_2 (double direction)		Δ_{dmp} , Δ_{d2mp}			
over	to	upper	low	upper	low
50	80	0	− 15	0	− 12
80	120	0	− 20	0	− 15
120	180	0	− 25	0	− 18
180	250	0	− 30	0	− 22
250	315	0	− 35	0	− 25
315	400	0	− 40	0	− 30
400	500	0	− 45	0	− 35
500	630	0	− 50	0	− 40
630	800	0	− 75	0	− 50
800	1000	0	− 100	0	
1000	1250	0	− 125	0	

Note: 1. d , Δ_{dmp} designate single direction parameters, d_2 , Δ_{d2mp} designate double direction parameters.

2. reference from GB/T 307. 4—2002.

Table 3 – 24 Thrust bearing housing washer diameter tolerance Δ_{Dmp}

(unit: μm)

Nominal housing washer bore diameter D /mm		grade 0, grade 6, grade 5		grade 4	
		Δ_{Dmp}			
over	to	upper	low	upper	low
10	18	0	− 11	0	− 7
18	30	0	− 13	0	− 8
30	50	0	− 16	0	− 9
50	80	0	− 19	0	− 11
80	120	0	− 22	0	− 13
120	180	0	− 25	0	− 15
180	250	0	− 30	0	− 20
250	315	0	− 35	0	− 25
315	400	0	− 40	0	− 28
400	500	0	− 45	0	− 33
500	630	0	− 50	0	− 38
630	800	0	− 75	0	− 45
800	1000	0	− 100	0	− 60
1000	1250	0	− 125	0	
1250	1600	0	− 160	0	

Note: Reference from GB/T 307. 4—2002.

(continued)

Basic dimension / mm		Shaft tolerance band													
		g6		g5		h6		h5		j5		j6		js6	
		clearance or interference													
over	to	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.
3	6	12	4	9	4	8	8	5	8	2	11	2	14	4	12
6	10	14	3	11	3	9	8	6	8	2	12	2	15	4.5	12.5
10	18	17	2	14	2	11	8	8	8	3	13	3	16	5.5	13.5
18	30	20	3	16	3	13	10	9	10	4	15	4	19	6.5	16.5
30	50	25	3	20	3	16	12	11	12	5	18	5	23	8	20
50	80	29	5	23	5	19	15	13	15	7	21	7	27	9.5	24.5
80	120	34	8	27	8	22	20	15	20	9	26	9	33	11	31
120 140 160	140 160 180	39	11	32	11	25	25	18	25	11	32	11	39	12.5	37.5
180 200 225	200 225 250	44	15	35	15	29	30	20	30	13	37	13	46	14.5	44.5
250 280	280 315	49	18	40	18	32	35	23	35	16	42	—	—	16	51
315 355	355 400	54	22	53	22	36	40	25	40	18	47	—	—	18	58
400 450	450 500	60	25	47	25	40	45	27	45	20	52	—	—	20	65

(continued)

Basic dimension / mm		Bearing bore diameter Δ _{dmp}		Shaft tolerance band															
				k5		k6		m5		m6		n6		p6		r6		r7	
				shaft diameter limit deviation															
over	to	upper	low																
3	6	0	− 8	+ 6	+ 1	+ 9	+ 1	+ 9	+ 4	+ 12	+ 4	+ 16	+ 8	+ 20	+ 12	—	—	—	—
6	10	0	− 8	+ 7	+ 1	+ 10	+ 1	+ 12	+ 6	+ 15	+ 6	+ 19	+ 10	+ 24	+ 15	—	—	—	—
10	18	0	− 8	+ 9	+ 1	+ 12	+ 1	+ 15	+ 7	+ 18	+ 7	+ 23	+ 12	+ 29	+ 18	—	—	—	—
18	30	0	− 10	+ 11	+ 2	+ 15	+ 2	+ 17	+ 8	+ 21	+ 8	+ 28	+ 15	+ 35	+ 22	—	—	—	—
30	50	0	− 12	+ 13	+ 2	+ 18	+ 2	+ 20	+ 9	+ 25	+ 9	+ 33	+ 17	+ 42	+ 26	—	—	—	—
50	80	0	− 15	+ 15	+ 2	+ 21	+ 2	+ 24	+ 11	+ 30	+ 11	+ 39	+ 20	+ 51	+ 32	—	—	—	—
80	120	0	− 20	+ 18	+ 3	+ 25	+ 3	+ 28	+ 13	+ 35	+ 13	+ 45	+ 23	+ 59	+ 37	—	—	—	—
120	140	0	− 25	+ 21	+ 3	+ 28	+ 3	+ 33	+ 15	+ 40	+ 15	+ 52	+ 27	+ 68	+ 43	+ 88	+ 63	—	—
140	160															+ 90	+ 65		
160	180															+ 93	+ 68		
180	200	0	− 30	+ 24	+ 4	+ 33	+ 4	+ 37	+ 17	+ 46	+ 17	+ 60	+ 31	+ 79	+ 50	+ 106	+ 77	+ 123	+ 77
200	225															+ 109	+ 80	+ 126	+ 80
225	250															+ 113	+ 84	+ 130	+ 84
250	280	0	− 35	+ 27	+ 4	+ 36	+ 4	+ 43	+ 20	+ 52	+ 20	+ 66	+ 34	+ 88	+ 56	+ 126	+ 94	+ 146	+ 94
280	315															+ 130	+ 98	+ 150	+ 98
315	355	0	− 40	+ 29	+ 4	+ 40	+ 4	+ 46	+ 21	+ 57	+ 21	+ 73	+ 37	+ 98	+ 62	+ 144	+ 108	+ 165	+ 108
355	400															+ 150	+ 114	+ 171	+ 114
400	450	0	− 45	+ 32	+ 5	+ 45	+ 5	+ 50	+ 23	+ 63	+ 23	+ 80	+ 40	+ 108	+ 68	+ 166	+ 126	+ 189	+ 126
450	500															+ 172	+ 132	+ 195	+ 132

(continued)

Basic dimension / mm		Shaft tolerance band															
		k5		k6		m5		m6		n6		p6		r6		r7	
		interference															
over	to	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
3	6	1	14	1	17	4	17	4	20	8	24	12	28	—	—	—	—
6	10	1	15	1	18	6	20	6	23	10	27	15	32	—	—	—	—
10	18	1	17	1	20	7	23	7	26	12	31	18	37	—	—	—	—
18	30	2	21	2	25	8	27	8	31	15	38	22	45	—	—	—	—
30	50	2	25	2	30	9	32	9	37	17	45	26	54	—	—	—	—
50	80	2	30	2	36	11	39	11	45	20	54	32	68	—	—	—	—
80	120	3	38	3	45	13	48	13	55	23	65	37	79	—	—	—	—
120 140 160	140 160 180	3	46	3	53	15	58	15	65	27	77	43	93	63 65 68	113 115 118	—	—
180 200 225	200 225 250	4	54	4	63	17	67	17	76	31	90	50	109	77 80 84	136 139 143	77 80 84	153 156 160
250 280	280 315	4	62	4	71	20	78	20	87	34	101	56	123	94 98	161 165	94 98	181 185
315 355	355 400	4	69	4	80	21	86	21	97	137	113	62	138	108 114	184 190	108 114	205 211
400 450	450 500	5	77	5	90	23	95	23	108	40	125	68	153	126 132	211 217	126 132	234 240

Table 3 - 26 Radial bearing (except tapered roller bearing) fitting to housing bore with the grade 0 tolerance (unit: μm)

Basic dimension /mm		Bearing outer diameter Δ_{Dmp}		Housing tolerance band															
				G7		H8		H7		H6		J7		J6		JS7		JS6	
				housing diameter limit deviation															
over	to	upper	low																
10	18	0	− 8	+ 24	+ 6	+ 27	0	+ 18	0	+ 11	0	+ 10	− 8	+ 6	− 5	+ 9	− 9	+ 5.5	− 5.5
18	30	0	− 9	+ 28	+ 7	+ 33	0	+ 21	0	+ 13	0	+ 12	− 9	+ 8	− 5	+ 10	− 10	+ 6.5	− 6.5
30	50	0	− 11	+ 34	+ 9	+ 39	0	+ 25	0	+ 16	0	+ 14	− 11	+ 10	− 6	+ 12	− 12	+ 8	− 8
50	80	0	− 13	+ 40	+ 10	+ 46	0	+ 30	0	+ 19	0	+ 18	− 12	+ 13	− 6	+ 15	− 15	+ 9.5	− 9.5
80	120	0	− 15	+ 47	+ 12	+ 54	0	+ 35	0	+ 22	0	+ 22	− 13	+ 16	− 6	+ 17	− 17	+ 11	− 11
120	150	0	− 18	+ 54	+ 14	+ 63	0	+ 40	0	+ 25	0	+ 26	− 14	+ 18	− 7	+ 20	− 20	+ 12.5	− 12.5
150	180	0	− 25	+ 54	+ 14	+ 63	0	+ 40	0	+ 25	0	+ 26	− 14	+ 18	− 7	+ 20	− 20	+ 12.5	− 12.5
180	250	0	− 30	+ 61	+ 15	+ 72	0	+ 46	0	+ 29	0	+ 30	− 16	+ 22	− 7	+ 23	− 23	+ 14.5	− 14.5
250	315	0	− 35	+ 69	+ 17	+ 81	0	+ 52	0	+ 32	0	+ 36	− 16	+ 25	− 7	+ 26	− 26	+ 16	− 16
315	400	0	− 40	+ 75	+ 18	+ 89	0	+ 57	0	+ 36	0	+ 39	− 18	+ 29	− 7	+ 28	− 28	+ 18	− 18
400	500	0	− 45	+ 83	+ 20	+ 97	0	+ 63	0	+ 40	+	+ 43	− 20	+ 33	− 7	+ 31	− 31	+ 20	− 20

(continued)

Basic dimension /mm		Housing bore tolerance band															
		G7		H8		H7		H6		J7		J7		JS7		JS6	
		clearance		clearance or interference													
over	to	max.	min.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.
10	18	32	6	35	0	26	0	19	0	18	8	14	5	17	9	13.5	5.5
18	30	37	7	42	0	30	0	22	0	21	9	17	5	19	10	15.5	6.5
30	50	45	9	50	0	36	0	27	0	25	11	21	6	23	12	19	8
50	80	53	10	59	0	43	0	32	0	31	12	26	6	28	15	22.5	9.5
80	120	62	12	69	0	50	0	37	0	37	13	31	6	32	17	26	11
120	150	72	14	81	0	58	0	43	0	44	14	36	7	38	20	30.5	12.5
150	180	79	14	88	0	65	0	50	0	51	14	43	7	45	20	37.5	12.5
180	250	91	15	102	0	76	0	59	0	60	16	52	7	53	23	44.5	14.5
250	315	104	17	116	0	87	0	67	0	71	16	60	7	61	26	51	16
315	400	115	18	129	0	97	0	76	0	79	18	69	7	68	28	58	18
400	500	128	20	142	0	108	0	85	0	88	20	78	7	76	31	65	20

(continued)

Basic dimension /mm		Bearing outer diameter Δ_{Dmp}		Housing tolerance band															
				K6		K7		M6		M7		N6		N7		P6		P7	
				housing diameter limit deviation															
over	to	upper	low																
10	18	0	− 8	+ 2	− 9	+ 6	− 12	− 4	− 15	0	− 18	− 9	− 20	− 5	− 23	− 15	− 26	− 11	− 29
18	30	0	− 9	+ 2	− 11	+ 6	− 15	− 4	− 17	0	− 21	− 11	− 24	− 7	− 28	− 18	− 31	− 14	− 35
30	50	0	− 11	+ 3	− 13	+ 7	− 18	− 4	− 20	0	− 25	− 12	− 28	− 8	− 33	− 21	− 37	− 17	− 42
50	80	0	− 13	+ 4	− 15	+ 9	− 21	− 5	− 24	0	− 30	− 14	− 33	− 9	− 39	− 26	− 45	− 21	− 51
80	120	0	− 15	+ 4	− 18	+ 10	− 25	− 6	− 28	0	− 35	− 16	− 38	− 10	− 45	− 30	− 52	− 24	− 59
120	150	0	− 18	+ 4	− 21	+ 12	− 28	− 8	− 33	0	− 40	− 20	− 45	− 12	− 52	− 36	− 61	− 28	− 68
150	180	0	− 25	+ 4	− 21	+ 12	− 28	− 8	− 33	0	− 40	− 20	− 45	− 12	− 52	− 36	− 61	− 28	− 68
180	250	0	− 30	+ 5	− 24	+ 13	− 33	− 8	− 37	0	− 46	− 22	− 51	− 14	− 60	− 41	− 70	− 33	− 79
250	315	0	− 35	+ 5	− 27	+ 16	− 36	− 9	− 41	0	− 52	− 25	− 57	− 14	− 66	− 47	− 79	− 36	− 88
315	400	0	− 40	+ 7	− 29	+ 17	− 40	− 10	− 46	0	− 57	− 26	− 62	− 16	− 73	− 51	− 87	− 41	− 98
400	500	0	− 45	+ 8	− 32	+ 18	− 45	− 10	− 50	0	− 63	− 27	− 67	− 17	− 80	− 55	− 95	− 45	− 108

(continued)

Basic dimension /mm		Housing bore tolerance band															
		K6		K7		M6		M7		N6		N7		P6		P7	
		clearance or interference												interference			
over	to	max. clear.	max. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	min.	max.	min.	max.
10	18	10	9	14	12	4	15	8	18	− 1	20	3	23	7	26	3	29
18	30	11	11	15	15	5	17	9	21	− 2	24	2	28	9	31	5	35
30	50	14	13	18	18	7	20	11	25	− 1	28	3	33	10	37	6	42
50	80	17	15	22	21	8	24	13	30	− 1	33	4	39	13	45	8	51
80	120	19	18	25	25	9	28	15	35	− 1	38	5	45	15	32	9	59
120	150	22	21	30	28	10	33	18	40	− 2	45	6	52	18	61	10	68
150	180	29	21	37	28	17	33	25	40	5	45	13	52	11	61	3	68
180	250	35	24	43	33	22	37	30	46	8	51	16	60	11	70	3	79
250	315	40	27	51	36	26	41	35	52	10	57	21	66	12	79	1	88
315	400	47	29	57	40	30	46	40	57	14	62	24	73	11	87	1	98
400	500	53	32	63	45	35	50	45	63	18	67	28	80	10	95	0	108

(continued)

Basic dimension /mm		Shaft tolerance band													
		g6		g5		h6		h5		j5		j6		js6	
		clearance or intereferece													
over	to	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.
3	6	12	3	9	3	8	7	5	7	2	10	2	13	4	11
6	10	14	2	11	2	9	7	6	7	2	11	2	14	4. 5	11. 5
10	18	17	1	14	1	11	7	8	7	3	12	3	15	5. 5	12. 5
18	30	20	1	16	1	13	8	9	8	4	13	4	17	6. 5	14. 5
30	50	25	1	20	1	16	10	11	10	5	16	5	21	8	18
50	80	29	2	23	2	19	12	13	12	7	18	7	24	9. 5	21. 5
80	120	34	3	27	3	22	15	15	15	9	21	9	28	11	26
120 140 160	140 160 180	39	4	32	4	25	18	18	18	11	25	11	32	12. 5	30. 5
180 200 225	200 225 250	44	7	35	7	29	22	20	22	13	29	13	38	14. 5	36. 5
250 280	280 315	49	8	40	8	32	25	23	25	16	32	—	—	16	41
315 355	355 400	54	12	43	12	36	30	25	30	18	37	—	—	18	48
400 450	450 500	60	15	47	15	40	35	27	35	20	42	—	—	20	55

(continued)

Basic dimension / mm		Bearing bore diameter Δ _{dmp}		Shaft tolerance band															
				k5		k6		m5		m6		n6		p6		r6		r7	
				shaft diameter limit deviation															
over	to	upper	low																
3	6	0	− 7	+ 6	+ 1	+ 9	+ 1	+ 9	+ 4	+ 12	+ 4	+ 16	+ 8	+ 20	+ 12	—	—	—	—
6	10	0	− 7	+ 7	+ 1	+ 10	+ 1	+ 12	+ 6	+ 15	+ 6	+ 19	+ 10	+ 24	+ 15	—	—	—	—
10	18	0	− 7	+ 9	+ 1	+ 12	+ 1	+ 15	+ 7	+ 18	+ 7	+ 23	+ 12	+ 29	+ 18	—	—	—	—
18	30	0	− 8	+ 11	+ 2	+ 15	+ 2	+ 17	+ 8	+ 21	+ 8	+ 28	+ 15	+ 35	+ 22	—	—	—	—
30	50	0	− 10	+ 13	+ 2	+ 18	+ 2	+ 20	+ 9	+ 25	+ 9	+ 33	+ 17	+ 42	+ 26	—	—	—	—
50	80	0	− 12	+ 15	+ 2	+ 21	+ 2	+ 24	+ 11	+ 30	+ 11	+ 39	+ 20	+ 51	+ 32	—	—	—	—
80	120	0	− 15	+ 18	+ 3	+ 25	+ 3	+ 28	+ 13	+ 35	+ 13	+ 45	+ 23	+ 59	+ 37	—	—	—	—
120	140	0	− 18	+ 21	+ 3	+ 28	+ 3	+ 33	+ 15	+ 40	+ 15	+ 52	+ 27	+ 68	+ 43	+ 88	+ 63	—	—
140	160															+ 90	+ 65		
160	180															+ 93	+ 68		
180	200	0	− 22	+ 24	+ 4	+ 33	+ 4	+ 37	+ 17	+ 46	+ 17	+ 60	+ 31	+ 79	+ 50	+ 106	+ 77	+ 123	+ 77
200	225															+ 109	+ 80	+ 126	+ 80
225	250															+ 113	+ 84	+ 130	+ 84
250	280	0	− 25	+ 27	+ 4	+ 36	+ 4	+ 43	+ 20	+ 52	+ 20	+ 66	+ 34	+ 88	+ 56	+ 126	+ 94	+ 146	+ 94
280	315															+ 130	+ 98	+ 150	+ 98
315	355															+ 144	+ 108	+ 165	+ 108
355	400	0	− 30	+ 29	+ 4	+ 40	+ 4	+ 46	+ 21	+ 57	+ 21	+ 73	+ 37	+ 98	+ 62	+ 150	+ 114	+ 171	+ 114
400	450															+ 166	+ 126	+ 189	+ 126
450	500															+ 172	+ 132	+ 195	+ 132

(continued)

Basic dimension /mm		Shaft tolerance band															
		k5		k6		m5		m6		n6		p6		r6		r7	
		interference															
over	to	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
3	6	1	13	1	16	4	16	4	19	8	23	12	27	—	—	—	—
6	10	1	14	1	17	6	19	6	22	10	26	15	31	—	—	—	—
10	18	1	16	1	19	7	22	7	25	12	30	18	36	—	—	—	—
18	30	2	19	2	23	8	25	8	29	15	36	22	43	—	—	—	—
30	50	2	23	2	28	9	30	9	35	17	43	26	52	—	—	—	—
50	80	2	27	2	33	11	36	11	42	20	51	32	63	—	—	—	—
80	120	3	33	3	40	13	43	13	50	23	60	37	74	—	—	—	—
120 140 160	140 160 180	3	39	3	46	15	51	15	58	27	70	43	86	63 65 68	106 108 111	—	—
180 200 225	200 225 250	4	46	4	55	17	59	17	68	31	82	50	101	77 80 84	128 131 135	77 80 84	145 148 152
250 280	280 315	4	52	4	61	20	68	20	77	34	91	58	113	94 98	151 155	94 98	171 175
315 355	355 400	4	59	4	70	21	76	21	87	37	103	62	128	108 114	174 180	108 114	195 201
400 450	450 500	5	67	5	80	23	85	23	98	40	115	68	143	126 132	201 207	126 132	224 230

Table 3-28 Radial bearing (except tapered roller bearing) fitting to housing bore with the grade 6 tolerance (unit: μm)

Basic dimension /mm		Bearing outer diameter Δ _{Dmp}		Housing tolerance band															
				G7		H8		H7		H6		J7		J6		JS7		JS6	
				housing diameter limit deviation															
over	to	upper	low																
10	18	0	− 7	+ 24	+ 6	+ 27	0	+ 18	0	+ 11	0	+ 10	− 8	+ 6	− 5	+ 9	− 9	+ 5.5	− 5.5
18	30	0	− 8	+ 28	+ 7	+ 33	0	+ 21	0	+ 13	0	+ 12	− 9	+ 8	− 5	+ 10	− 10	+ 6.5	− 6.5
30	50	0	− 9	+ 34	+ 9	+ 39	0	+ 25	0	+ 16	0	+ 14	− 11	+ 10	− 6	+ 12	− 12	+ 8	− 8
50	80	0	− 11	+ 40	+ 10	+ 46	0	+ 30	0	+ 19	0	+ 18	− 12	+ 13	− 6	+ 15	− 15	+ 9.5	− 9.5
80	120	0	− 13	+ 47	+ 12	+ 54	0	+ 35	0	+ 22	0	+ 22	− 13	+ 16	− 6	+ 17	− 17	+ 11	− 11
120	150	0	− 15	+ 54	+ 14	+ 63	0	+ 40	0	+ 25	0	+ 26	− 14	+ 18	− 7	+ 20	− 20	+ 12.5	− 12.5
150	180	0	− 18	+ 54	+ 14	+ 63	0	+ 40	0	+ 25	0	+ 26	− 14	+ 18	− 7	+ 20	− 20	+ 12.5	− 12.5
180	250	0	− 20	+ 61	+ 15	+ 72	0	+ 46	0	+ 29	0	+ 30	− 16	+ 22	− 7	+ 23	− 23	+ 14.5	− 14.5
250	315	0	− 25	+ 69	+ 17	+ 81	0	+ 52	0	+ 32	0	+ 36	− 16	+ 25	− 7	+ 26	− 26	+ 16	− 16
315	400	0	− 28	+ 75	+ 18	+ 89	0	+ 57	0	+ 36	0	+ 39	− 18	+ 29	− 7	+ 28	− 28	+ 18	− 18
400	500	0	− 33	+ 83	+ 20	+ 97	0	+ 63	0	+ 40	+	+ 43	− 20	+ 33	− 7	+ 31	− 31	+ 20	− 20

(continued)

Basic dimension /mm		Housing bore tolerance band															
		G7		H8		H7		H6		J7		J7		JS7		JS6	
		clearance		clearance or interference													
over	to	max.	min.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.
10	18	31	6	34	0	25	0	18	0	17	8	13	5	16	9	12.5	5.5
18	30	36	7	41	0	29	0	21	0	20	9	16	5	18	10	14.5	6.5
30	50	43	9	48	0	34	0	25	0	23	11	19	6	21	12	17	8
50	80	51	10	57	0	41	0	30	0	29	12	24	6	26	15	20.5	9.5
80	120	60	12	67	0	48	0	35	0	35	13	29	6	30	17	24	11
120	150	69	14	78	0	55	0	40	0	41	14	33	7	35	20	27.5	12.5
150	180	72	14	81	0	58	0	43	0	44	14	36	7	38	20	30.5	12.5
180	250	81	15	92	0	66	0	49	0	50	16	42	7	43	23	34.5	14.5
250	315	94	17	106	0	77	0	57	0	61	16	50	7	51	26	41	16
315	400	103	18	117	0	85	0	64	0	67	18	57	7	56	28	46	18
400	500	116	20	130	0	96	0	73	0	76	20	66	7	64	31	53	20

(continued)

Basic dimension /mm		Bearing outer diameter Δ _{Dmp}		Housing tolerance band															
				K6		K7		M6		M7		N6		N7		P6		P7	
				housing diameter limit deviation															
over	to	upper	low																
10	18	0	− 7	+ 2	− 9	+ 6	− 12	− 4	− 15	0	− 18	− 9	− 20	− 5	− 23	− 15	− 26	− 11	− 29
18	30	0	− 8	+ 2	− 11	+ 6	− 15	− 4	− 17	0	− 21	− 11	− 24	− 7	− 28	− 18	− 31	− 14	− 35
30	50	0	− 9	+ 3	− 13	+ 7	− 18	− 4	− 20	0	− 28	− 12	− 28	− 8	− 33	− 21	− 37	− 17	− 42
50	80	0	− 11	+ 4	− 15	+ 9	− 21	− 5	− 24	0	− 30	− 14	− 33	− 9	− 39	− 26	− 45	− 21	− 51
80	120	0	− 13	+ 4	− 18	+ 10	− 25	− 6	− 28	0	− 35	− 16	− 38	− 10	− 45	− 30	− 52	− 24	− 59
120	150	0	− 15	+ 4	− 21	+ 12	− 28	− 8	− 33	0	− 40	− 20	− 45	− 12	− 52	− 36	− 61	− 28	− 68
150	180	0	− 18	+ 4	− 21	+ 12	− 28	− 8	− 33	0	− 40	− 20	− 45	− 12	− 52	− 36	− 61	− 28	− 68
180	250	0	− 20	+ 5	− 24	+ 13	− 33	− 8	− 37	0	− 46	− 22	− 51	− 14	− 60	− 41	− 70	− 33	− 79
250	315	0	− 25	+ 5	− 27	+ 16	− 36	− 9	− 41	0	− 52	− 25	− 57	− 14	− 66	− 47	− 79	− 36	− 88
315	400	0	− 28	+ 7	− 29	+ 17	− 40	− 10	− 46	0	− 57	− 26	− 62	− 16	− 73	− 51	− 87	− 41	− 98
400	500	0	− 33	+ 8	− 32	+ 18	− 45	− 10	− 50	0	− 63	− 27	− 67	− 17	− 80	− 55	− 95	− 45	− 108

(continued)

Basic dimension /mm		Housing bore tolerance band															
		K6		K7		M6		M7		N6		N7		P6		P7	
		clearance or interference												interference			
over	to	max. clear.	max. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	max. clear.	min. inter.	min.	max.	min.	max.
10	18	9	9	13	12	3	15	7	18	− 2	20	2	23	8	26	4	29
18	30	10	11	14	15	4	17	8	21	− 3	24	1	28	10	31	6	35
30	50	12	13	16	18	5	20	9	25	− 3	28	1	33	12	37	8	42
50	80	15	15	20	21	6	24	11	30	− 3	33	2	39	15	45	10	51
80	120	17	18	23	25	7	28	13	35	− 3	38	3	45	17	52	11	59
120	150	19	21	27	28	7	33	15	40	− 5	45	3	52	21	61	13	68
150	180	22	21	30	28	10	33	18	40	− 2	45	6	52	18	61	10	68
180	250	25	24	33	33	12	37	20	46	− 2	51	6	60	21	70	13	79
250	315	30	27	41	36	16	41	25	52	0	57	11	66	22	79	11	88
315	400	35	29	45	40	18	46	28	57	2	62	12	73	23	87	13	98
400	500	41	32	51	45	23	50	33	63	6	67	16	80	22	95	12	108

Table 3-29 Tapered roller bearing (grade 0, 6x) fitting to shaft

 $(\text{unit: } \mu\text{m})$

Basic dimension /mm		Bearing bore diameter Δ_{dmp}		Shaft tolerance band													
				f6		g6		g5		h6		h5		j5		j6	
				shaft diameter limit deviation													
over	to	upper	low														
10	18	0	−12	−16	−27	−6	−17	−6	−14	0	−11	0	−8	+5	−3	+8	−3
18	30	0	−12	−20	−33	−7	−20	−7	−16	0	−13	0	−9	+5	−4	+9	−4
30	50	0	−12	−25	−41	−9	−25	−9	−20	0	−16	0	−11	+6	−5	+11	−5
50	80	0	−15	−30	−49	−10	−29	−10	−23	0	−19	0	−13	+6	−7	+12	−7
80	120	0	−20	−36	−58	−12	−34	−12	−27	0	−22	0	−15	+6	−9	+13	−9
120	140	0	−25	−43	−68	−14	−39	−14	−32	0	−25	0	−18	+7	−11	+14	−11
140	160																
160	180																
180	200	0	−30	−50	−79	−15	−44	−15	−35	0	−29	0	−20	+7	−13	+16	−13
200	225																
225	250																
250	280	0	−35	−56	−88	−17	−49	−17	−40	0	−32	0	−23	+7	−16	—	—
280	315																
315	355	0	−40	−62	−98	−18	−54	−18	−43	0	−36	0	−25	+7	−18	—	—
355	400																
400	450	0	−45	−68	−108	−20	−60	−20	−47	0	−40	0	−27	+7	−20	—	—
450	500																

(continued)

Basic dimension /mm		Shaft tolerance band													
		f6		g6		g5		h6		h5		j5		j6	
		clearance				clearance or intereferece									
over	to	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.
10	18	27	4	17	6	14	6	11	12	8	12	3	18	3	20
18	30	33	8	20	5	16	5	13	12	9	12	4	18	4	21
30	50	41	13	25	3	20	3	16	12	11	12	5	18	5	23
50	80	49	15	29	5	23	5	19	15	13	15	7	21	7	27
80	120	58	16	34	8	27	8	22	20	15	20	9	26	9	33
120 140 160	140 160 180	28	18	39	11	32	11	25	25	18	25	11	32	11	39
180 200 225	200 225 250	79	20	44	15	35	15	29	30	20	30	13	37	13	46
250 280	280 315	88	21	49	18	40	18	32	35	23	35	16	42	—	—
315 355	355 400	98	22	54	22	43	22	36	40	25	40	18	47	—	—
400 450	450 500	108	23	60	25	47	25	40	45	27	45	20	52	—	—

(continued)

Basic dimension /mm		Bearing bore diameter Δ _{dmp}		Shaft tolerance band															
				js6		k5		k6		m5		m6		n6		p6		r6	
				shaft diameter limit deviation															
over	to	upper	low																
10	18	0	−12	+5.5	−5.5	+9	+1	+12	+1	+15	+7	+18	+7	+23	+12	+29	+18	—	—
18	30	0	−12	+6.5	−6.5	+11	+2	+15	+2	+17	+8	+21	+8	+28	+15	+35	+22	—	—
30	50	0	−12	+8	−8	+13	+2	+18	+2	+20	+9	+25	+9	+33	+17	+42	+26	—	—
50	80	0	−15	+9.5	−9.5	+15	+2	+21	+2	+24	+11	+30	+11	+39	+20	+51	+32	—	—
80	120	0	−20	+11	−11	+18	+3	+25	+3	+28	+13	+35	+13	+45	+23	+59	+37	—	—
120	140	0	−25	+12.5	−12.5	+21	+3	+28	+3	+33	+15	+40	+15	+52	+27	+68	+43	+68	+63
140	160																	+90	+65
160	180																	+93	+68
180	200	0	−30	+14.5	−14.5	+24	+4	+33	+4	+37	+17	+46	+17	+60	+31	+79	+50	+106	+77
200	225																	+109	+80
225	250																	+113	+84
250	280	0	−35	+16	−16	+27	+4	+36	+4	+43	+20	+52	+20	+66	+34	+88	+56	+126	+94
280	315																	+130	+98
315	355																	+144	+108
355	400	0	−40	+18	−18	+29	+4	+40	+4	+46	+21	+57	+21	+73	+37	+98	+62	+150	+114
400	450	0	−45	+20	−20	+3	+5	+45	+5	+50	+23	+63	+23	+80	+40	+108	+68	+166	+126
450	500																	+172	+132

(continued)

Basic dimension /mm		Shaft tolerance band															
		js6		k5		k6		m5		m6		n6		p6		r6	
		clearance or inter ference				inter ference											
over	to	max. clear.	max. inter.	max. clear.	max. inter.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
10	18	5.5	17.5	—	—	—	—	—	—	—	—	—	—	—	—	—	—
18	30	6.5	18.5	2	23	2	27	—	—	—	—	—	—	—	—	—	—
30	50	8	20	2	25	2	30	9	32	9	37	—	—	—	—	—	—
50	80	9.5	24.5	2	30	2	36	11	39	11	45	20	54	—	—	—	—
80	120	11	31	3	38	3	45	13	48	13	55	23	65	37	79	—	—
120 140 160	140 160 180	12.5	37.5	3	46	3	53	15	58	15	65	27	77	43	93	63 65 68	113 115 118
180 200 225	200 225 250	14.5	44.5	4	54	4	63	17	67	17	67	31	90	50	109	77 80 84	136 139 143
250 280	280 315	16	51	4	62	4	71	20	78	20	87	34	101	56	123	94 98	161 165
315 355	355 400	18	58	4	69	4	80	21	86	21	97	37	113	62	138	108 114	184 190
400 450	450 500	20	65	5	77	5	90	23	95	23	108	40	125	68	153	126 132	211 217

Table 3 – 30 Tapered roller bearing (grade 0, 6x) fitting to housing bore(unit: μm)

Basic dimension /mm		Bearing outer diameter Δ_{Dmp}		Housing tolerance band															
				G7		H8		H7		H6		J7		J6		JS7		JS6	
				housing bore diameter limit deviation															
over	to	upper	low																
30	50	0	− 14	+ 34	+ 9	+ 39	0	+ 25	0	+ 16	0	+ 14	− 11	+ 10	− 6	+ 12	− 12	+ 8.5	− 8.5
50	80	0	− 16	+ 40	+ 10	+ 46	0	+ 30	0	+ 19	0	+ 18	− 12	+ 13	− 6	+ 15	− 15	+ 9.5	− 9.5
80	120	0	− 18	+ 47	+ 12	+ 54	0	+ 35	0	+ 22	0	+ 22	− 13	+ 16	− 6	+ 17	− 17	+ 11	− 11
120	150	0	− 20	+ 54	+ 14	+ 63	0	+ 40	0	+ 25	0	+ 26	− 14	+ 18	− 7	+ 20	− 20	+ 12.5	− 12.5
150	180	0	− 25	+ 54	+ 14	+ 63	0	+ 40	0	+ 25	0	+ 26	− 14	+ 18	− 7	+ 20	− 20	+ 12.5	− 12.5
180	250	0	− 30	+ 61	+ 15	+ 72	0	+ 46	0	+ 29	0	+ 30	− 16	+ 22	− 7	+ 23	− 23	+ 14.5	− 14.5
250	315	0	− 35	+ 69	+ 17	+ 81	0	+ 52	0	+ 32	0	+ 36	− 16	+ 25	− 7	+ 26	− 26	+ 16	− 16
315	400	0	− 40	+ 75	+ 18	+ 89	0	+ 57	0	+ 36	0	+ 39	− 18	+ 29	− 7	+ 28	− 28	+ 18	− 18
400	500	0	− 45	+ 83	+ 20	+ 97	0	+ 63	0	+ 40	0	+ 43	− 20	+ 33	− 7	+ 31	− 31	+ 20	− 20

(continued)

Basic dimension /mm		Housing bore tolerance band															
		G7		H8		H7		H6		J7		J6		JS7		JS6	
		clearance		clearance or interference													
over	to	max.	min.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.
30	50	48	9	50	0	39	0	30	0	28	11	24	6	26	12	22	8
50	80	56	10	59	0	46	0	35	0	34	12	29	6	31	15	25.5	9.5
80	120	65	12	69	0	53	0	40	0	40	13	34	6	35	17	29	11
120	150	74	14	81	0	60	0	45	0	46	14	38	7	40	20	32.5	12.5
150	180	79	14	88	0	65	0	50	0	51	14	43	7	45	20	37.5	12.5
180	250	91	15	102	0	76	0	59	0	60	16	52	7	53	23	44.5	14.5
250	315	104	17	116	0	87	0	67	0	71	16	60	7	61	26	51	16
315	400	115	18	129	0	97	0	76	0	79	18	69	7	68	28	58	18
400	500	128	20	142	0	108	0	85	0	88	20	78	7	76	31	65	20

(continued)

Basic dimension /mm		Bearing outer diameter Δ_{Dmp}		Housing tolerance band															
				K6		K7		M6		M7		N6		N7		P6		P7	
				housing diameter limit deviation															
over	to	upper	low																
30	50	0	− 11	+ 3	− 13	+ 7	− 18	− 4	− 20	0	− 25	− 12	− 28	− 8	− 33	− 21	− 37	− 17	− 42
50	80	0	− 13	+ 4	− 15	+ 9	− 21	− 5	− 24	0	− 30	− 14	− 33	− 9	− 39	− 26	− 45	− 21	− 51
80	120	0	− 15	+ 4	− 18	+ 10	− 25	− 6	− 28	0	− 35	− 16	− 38	− 10	− 45	− 30	− 52	− 24	− 59
120	150	0	− 18	+ 4	− 21	+ 12	− 28	− 8	− 33	0	− 40	− 20	− 45	− 12	− 52	− 36	− 61	− 28	− 68
150	180	0	− 25	+ 4	− 21	+ 12	− 28	− 8	− 33	0	− 40	− 20	− 45	− 12	− 52	− 36	− 61	− 28	− 68
180	250	0	− 30	+ 5	− 24	+ 13	− 33	− 8	− 37	0	− 46	− 22	− 51	− 14	− 60	− 41	− 70	− 33	− 79
250	315	0	− 35	+ 5	− 27	+ 16	− 36	− 9	− 41	0	− 52	− 25	− 57	− 14	− 66	− 47	− 79	− 36	− 88
315	400	0	− 40	+ 7	− 29	+ 17	− 40	− 10	− 46	0	− 57	− 26	− 62	− 16	− 73	− 51	− 87	− 41	− 98
400	500	0	− 45	+ 8	− 32	+ 18	− 45	− 10	− 50	0	− 63	− 27	− 67	− 17	− 80	− 55	− 95	− 45	− 108

(continued)

Basic dimension /mm		Housing bore tolerance band															
		K6		K7		M6		M7		N6		N7		P6		P7	
		clearance or interference												interference			
over	to	max. clear.	min. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	min.	max.	min.	max.
30	50	17	13	21	18	10	20	14	25	2	28	6	33	7	37	3	42
50	80	20	15	25	21	11	24	16	30	2	33	7	39	10	45	5	51
80	120	22	18	28	25	12	28	18	35	2	38	8	45	12	52	6	59
120	150	24	21	32	28	12	33	20	40	0	45	8	52	16	61	8	68
150	180	29	21	37	28	17	33	25	40	5	45	13	52	11	61	3	68
180	250	35	24	43	33	22	37	30	46	8	51	16	60	11	70	3	79
250	315	40	27	51	36	26	41	35	52	10	57	21	66	12	79	1	88
315	400	47	29	57	40	30	46	40	57	14	62	24	73	11	87	1	98
400	500	53	32	63	45	35	50	45	63	18	67	28	80	10	95	0	108

Table 3-31 grade 5 bearing fitting to shaft

(unit: μm)

Basic dimension /mm		Bearing bore diameter $\Delta_{\text{dmp}}^{\text{①}}$		Shaft tolerance band													
				h5		j5		js5		k5		k6		m5		m6	
				shaft diameter limit deviation													
over	to	upper	low														
3	6	0	−5	0	−5	+3	−2	+2.5	−2.5	+6	+1	+9	+1	+9	+4	+12	+4
6	10	0	−5	0	−6	+4	−2	+3	−3	+7	+1	+10	+1	+12	+6	+15	+6
10	18	0	−5	0	−8	+5	−3	+4	−4	+9	+1	+12	+1	+15	+7	+18	+7
18	30	0	−6	0	−9	+5	−4	+4.5	−4.5	+11	+2	+15	+2	+17	+8	+21	+8
30	50	0	−8	0	−11	+6	−5	+5.5	−5.5	+13	+2	+18	+2	+20	+9	+25	+9
50	80	0	−9	0	−13	+6	−7	+6.5	−6.5	+15	+2	+21	+2	+24	+11	+30	+11
80	120	0	−10	0	−15	+6	−9	+7.5	−7.5	+18	+3	+25	+3	+28	+13	+35	+13
120	180	0	−13	0	−18	+7	−11	+9	−9	+21	+3	+28	+3	+33	+15	+40	+15
180	250	0	−15	0	−20	+7	−13	+10	−10	+24	+4	+35	+4	+37	+17	+46	+17
250	315	0	−18	0	−23	+7	−16	+11.5	−11.5	+27	+4	+36	+4	+43	+20	+52	+20
315	400	0	−23	0	−25	+7	−18	+12.5	−12.5	+29	+4	+40	+4	+46	+21	+57	+21

(continued)

Basic dimension /mm		Shaft tolerance band													
		h5		j5		js5		k5		k6		m5		m6	
		clearance or interference						interference							
over	to	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	min.	max.	min.	max.	min.	max.	min.	max.
3	6	5	5	2	8	2.5	7.5	1	11	1	14	4	14	4	17
6	10	6	5	2	9	3	8	1	12	1	15	6	17	6	20
10	18	8	5	3	10	4	9	1	14	1	17	7	20	7	23
18	30	9	6	4	11	4.5	10.5	2	17	2	21	8	23	8	27
30	50	11	8	5	14	5.5	13.5	2	21	2	26	9	28	9	33
50	80	13	9	7	15	6.5	15.5	2	24	2	30	11	33	11	39
80	120	15	10	9	16	7.5	17.5	3	28	3	35	13	38	13	45
120	180	18	13	11	20	9	22	3	34	3	41	15	46	15	53
180	250	20	15	13	22	10	25	4	39	4	48	17	52	17	61
250	315	23	18	16	25	11.5	29.5	4	45	4	54	20	61	20	70
315	400	25	23	18	30	12.5	35.5	4	52	4	63	21	69	21	80

Note: ① Δ_{dmp} — bearing inner ring single plane even bore diameter tolerance.

Table 3-32 grade 5 bearing fitting to housing bore

(unit: μm)

Basic dimension /mm		Bearing outer diameter $\Delta_{\text{Dmp}}^{\text{①}}$		Housing tolerance band															
				G6		H6		JS5		JS6		K5		K6		M5		M6	
over	to	upper	low	housing diameter limit deviation															
10	18	0	-5	+17	+6	+11	0	+4	-4	+5.5	-5.5	+2	-6	+2	-9	-4	-12	-4	-15
18	30	0	-6	+20	+7	+13	0	+4.5	-4.5	+6.5	-6.5	+1	-8	+2	-11	-5	-14	-4	-17
30	50	0	-7	+25	+9	+16	0	+5.5	-5.5	+8	-8	+2	-9	+3	-13	-5	-16	-4	-20
50	80	0	-9	+29	+10	+19	0	+6.5	-6.5	+9.5	-9.5	+3	-10	+4	-15	-6	-19	-5	-24
80	120	0	-10	+34	+12	+22	0	+7.5	-7.5	+11	-11	+2	-13	+4	-18	-8	-23	-6	-28
120	150	0	-11	+39	+14	+25	0	+9	-9	+12.5	-12.5	+3	-15	+4	-21	-9	-27	-8	-33
150	180	0	-13	+39	+14	+25	0	+9	-9	+12.5	-12.5	+3	-15	+4	-21	-9	-27	-8	-33
180	250	0	-15	+44	+15	+29	0	+10	-10	+14.5	-14.5	+2	-18	+5	-24	-11	-31	-8	-37
250	315	0	-18	+49	+17	+32	0	+11.5	-11.5	+16	-16	+3	-20	+5	-27	-13	-36	-9	-41
315	400	0	-20	+54	+18	+36	0	+12.5	-12.5	+18	-18	+3	-22	+7	-29	-14	-39	-10	-46
400	500	0	-23	+60	+20	+40	0	+13.5	-13.5	+20	-20	+2	-25	+8	-32	-16	-43	-10	-50

(continued)

Basic dimension /mm		Clearance		Clearance or interference													
		G6		H6		JS5		JS6		K5		K6		M5		M6	
		housing diameter limit deviation															
over	to	max.	min.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.
10	18	22	6	16	0	9	4	10.5	5.5	7	6	7	9	1	12	1	15
18	30	26	7	19	0	10.5	4.5	12.5	6.5	7	8	8	11	1	14	2	17
30	50	32	9	23	0	12.5	5.5	15	8	9	9	10	13	2	16	3	20
50	80	38	10	28	0	15.5	6.5	18.5	9.5	12	10	13	15	3	19	4	24
80	120	44	12	32	0	17.5	7.5	21	11	12	13	14	18	2	23	4	28
120	150	50	14	36	0	20	9	23.5	12.5	14	15	15	21	2	27	3	33
150	180	52	14	38	0	22	9	25.5	12.5	16	15	17	21	4	27	5	33
180	250	59	15	44	0	25	10	29.5	14.5	17	18	20	24	4	31	7	37
250	315	67	17	50	0	29.5	11.5	34	16	21	20	23	27	5	36	9	41
315	400	74	18	56	0	32.5	12.5	38	18	23	22	27	29	6	39	10	46
400	500	83	20	63	0	36.5	13.5	43	20	25	25	31	32	7	43	13	50

Note: ① Δ_{Dmp} — bearing outer ring single plane even outer diameter tolerance.

Table 3-33 grade 4 bearing fitting to shaft

(unit: μm)

Basic dimension /mm		Bearing outer diameter $\Delta_{\text{dmp}}^{\text{①}}$		Shaft tolerance band													
				h4		h5		js4		js5		k4		k5		m5	
over	to	upper	low	shaft diameter limit deviation													
3	6	0	-4	0	-4	0	-5	+2	-2	+2.5	-2.5	+5	+1	+6	+1	+9	+4
6	10	0	-4	0	-4	0	-6	+2	-2	+3	-3	+5	+1	+7	+1	+12	+6
10	18	0	-4	0	-5	0	-8	+2.5	-2.5	+4	-4	+6	+1	+9	+1	+15	+7
18	30	0	-5	0	-6	0	-9	+3	-3	+4.5	-4.5	+8	+2	+11	+2	+17	+8
30	50	0	-6	0	-7	0	-11	+3.5	-3.5	+5.5	-5.5	+9	+2	+13	+2	+20	+9
50	80	0	-7	0	-8	0	-13	+4	-4	+6.5	-6.5	+10	+2	+15	+2	+24	+11
80	120	0	-8	0	-10	0	-15	+5	-5	+7.5	-7.5	+13	+3	+18	+3	+28	+13
120	180	0	-10	0	-12	0	-18	+6	-6	+9	-9	+15	+3	+21	+3	+33	+15
180	250	0	-12	0	-14	0	-20	+7	-7	+10	-10	+18	+4	+24	+4	+37	+17
Basic dimension /mm				clearance or interference								interference					
				h4		h5		js4		js5		k4		k5		m5	
over	to	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	min.	max.	min.	max.	min.	max.		
3	6	4	4	5	4	2	6	2.5	6.5	1	9	1	10	4	13		
6	10	4	4	6	4	2	6	3	7	1	9	1	11	6	16		
10	18	5	4	8	4	2.5	6.5	4	8	1	10	1	13	7	19		
18	30	6	5	9	5	3	8	4.5	9.5	2	13	2	16	8	22		
30	50	7	6	11	6	3.5	9.5	5.5	11.5	2	15	2	19	9	26		
50	80	8	7	13	7	4	11	6.5	13.5	2	17	2	22	11	31		
80	120	10	8	15	8	5	13	7.5	15.5	3	21	3	26	13	36		
120	180	12	10	18	10	6	16	9	19	3	25	3	31	15	43		
180	250	14	12	20	12	7	19	10	22	4	30	4	36	17	49		

Note: ① Δ_{dmp} —bearing inner ring single plane even bore diameter tolerance.

Table 3 – 34 grade 4 bearing fitting to housing bore (unit: μm)

Basic dimension /mm		$\Delta_{\text{Dmp}}^{\text{①}}$		Housing tolerance band																			
				H5		JS5		K5		K6		M5											
over	to	upper	low	housing diameter limit deviation																			
10	18	0	-4	-8	0	+4	-4	+2	-6	+2	-9	-4	-12										
18	30	0	-5	+9	0	+4.5	-4.5	+1	-8	+2	-11	-5	-14										
30	50	0	-6	+11	0	+5.5	-5.5	+2	-9	+3	-13	-5	-16										
50	80	0	-7	+13	0	+6.5	-6.5	+3	-10	+4	-15	-6	-19										
80	120	0	-8	+15	0	+7.5	-7.5	+2	-13	+4	-18	-8	-23										
120	150	0	-9	+18	0	+9	-9	+3	-15	+4	-21	-9	-27										
150	180	0	-10	+18	0	+9	-9	+3	-15	+4	-21	-9	-27										
180	250	0	-11	+20	0	+10	-10	+2	-18	+5	-24	-11	-31										
250	315	0	-13	+23	0	+11.5	-11.5	+3	-20	+5	-27	-13	-36										
315	400	0	-15	+25	0	+12.5	-12.5	+3	-22	+7	-29	-14	-39										
basic dimension /mm		$\Delta_{\text{Dmp}}^{\text{①}}$		housing tolerance band																			
				H5		JS5		K5		K6		M5											
basic dimension/mm														clearance or interference									
over		to		max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.										
10		18		12	0	8	4	6	6	6	9	0	12										
18		30		14	0	9.5	4.5	6	8	7	11	0	14										
30		50		17	0	11.5	5.5	8	9	9	13	1	16										
50		80		20	0	13.5	6.5	10	10	11	15	1	19										
80		120		23	0	15.5	7.5	10	13	12	18	0	23										
120		150		27	0	18	9	12	15	13	21	0	27										
150		180		28	0	19	9	13	15	14	21	1	27										
180		250		31	0	21	10	13	18	16	24	0	31										
250		315		36	0	24.5	11.5	16	20	18	27	0	36										
315		400		40	0	27.5	12.5	18	22	22	29	1	39										

Note: ① Δ_{Dmp} —bearing outer ring single plane even diameter tolerance.

Table 3 – 35 grade 2 bearing fitting to shaft (unit: μm)

Basic dimension/mm		Bearing bore diameter Δ_{dmp}		fit			
				js3		h3	
				shaft diameter limit deviation			
over ^①	to	upper	low	upper	low	upper	low
0.6	2.5	0	-2.5	+1	-1	0	-2
2.5	6		-2.5	+1.25	-1.25		-2.5
6	10		-2.5	+1.25	-1.25		-2.5
10	18		-2.5	+1.5	-1.5		-3
18	30		-2.5	+2	-2		-4
30	50		-2.5	+2	-2		-4
50	80		-4	+2.5	-2.5		-5
80	120		-5	+3	-3		-6
120	150		-7	+4	-4		-8
150	180		-7	+4	-4		-8
180	250	-8	+5	-5	-10		

nominal diamele/mm		clearance or interference			
over ^①	to	max. clear.	max. inter.	max. clear.	max. inter.
0.6	2.5	3.5	1	2.5	2
2.5	6	3.75	1.25	2.5	2.5
6	10	3.75	1.25	2.5	2.5
10	18	4	1.5	2.5	3
18	30	4.5	2	2.5	4
30	50	4.5	2	2.5	4
50	80	6.5	2.5	4	5
80	120	8	3	5	6
120	150	11	4	7	8
150	180	11	4	7	8
180	250	13	5	8	10

① Include 0.6.

Table 3 - 36 grade 2 bearing fitting to housing bore

(unit: μm)

Nominal bearing diameter/mm		Δ_{Dmp}		fit									
				K4		JS3		JS4		H3		H4	
				housing bore diameter limit deviation									
over ^①	to	upper	low	upper	low	upper	low	upper	low	upper	low	upper	low
2.5	6	0	-2.5	—	—	+1.25	-1.25	+2	-2	+2.5	0	+4	0
6	10		-2.5	—	—	+1.25	-1.25	+2	-2	+2.5		+4	
10	18		-2.5	—	—	+1.5	-1.5	+2.5	-2.5	+3		+5	
18	30		-4	0	-6	+2	-2	+3	-3	+4		+6	
30	50		-4	+1	-6	+2	-2	+3.5	-3.5	+4		+7	
50	80		-4	+1	-7	+2.5	-2.5	+4	-4	+5		+8	
80	120		-5	+1	-9	+3	-3	+5	-5	+6		+10	
120	150		-5	+1	-11	+4	-4	+6	-6	+8		+12	
150	180		-7	+1	-11	+4	-4	+6	-6	+8		+12	
180	250		-8	0	-14	+5	-5	+7	-7	+10		+14	
250	315		-8	0	-16	+6	-6	+8	-8	+12		+16	
315	400		-10	+1	-17	+6.5	-6.5	+9	-9	+13		+18	

(continued)

Nominal bearing diameter /mm				Interference and clearance							
over ^①	to	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.	max. clear.	max. inter.
2.5	6			3.75	1.25	4.5	2	5		6.5	
6	10			3.75	1.25	4.5	2	5		6.5	
10	18			4	1.5	5	2.5	5.5		7.5	
18	30	4	6	6	2	7	3	8		10	
30	50	5	6	6	2	7.5	3.5	8	0	11	0
50	80	5	7	6.5	2.5	8	4	9		12	
80	120	6	9	8	3	10	5	11		15	
120	150	6	11	9	4	11	6	13		17	
150	180	8	11	11	4	13	6	15		19	
180	250	8	14	13	5	15	7	18		22	
250	315	8	16	14	6	16	8	20		24	
315	400	11	17	16.5	6.5	19	9	23		28	

① include 2.5.

3.3 Manufacturing Accuracy of Matching Components with Rolling Bearings

The manufacturing accuracy of matching shaft and housing bore should include dimension accuracy, geometric tolerance and surface roughness. The bearing accuracy grades are from the lowest grade 0 to grade 2, and their average diameter tolerances correspond to IT7 to IT3. The shaft diameter tolerance matching with grade 0 bearing is generally correspondent to IT6; the housing is generally correspondent to IT7. The cylindricity tolerand should be one or two IT grades higher than the dimension tolerance grade. For example, if taking the shaft diameter tolerance as M6, the geometric tolerance grade should be IT5 or IT4, then the cylindricity tolerance should be IT5/2 or IT4/2. If using adapter sleeve or withdrawal sleeve, it is permissible to adopt a little wider shaft diameter tolerance grade, such as IT8 or IT9, but the requirement for the geometric tolerance grade of the matching shaft diameter is strict.

The cylindricity tolerance should be IT6/2 (for h8) or IT7 (for h9). Next, we will introduce the value methods for the geometric tolerance and surface roughness of the matching shaft and matching housing bore, and the marking method for geometric tolerance.

Table 3 – 37 lists the geometric tolerance of the shaft diameter and housing bore matching to grade 0 and grade 6 bearings; Table 3 – 38 lists the surface roughness for matching surfaces. Figure 3 – 9 and Figure 3 – 10 are the marking methods for geometric tolerance.

Table 3 – 37 Geometric tolerance of shaft diameter and housing bore

Basic dimension /mm		Deviation from Cylindrical form t				End face circle runout t_1			
		shaft diameter		housing bore		shaft abutment		housing bore rib	
		bearing accuracy grade							
		0	6(6X)	0	6(6X)	0	6(6X)	0	6(6X)
over	to			tolerance value/ μm					
	6	2.5	1.5	4	2.5	5	3	8	5
6	10	2.5	1.5	4	2.5	6	4	10	6
10	18	3.0	2.0	5	3.0	8	5	12	8

(continued)

Basic dimension /mm		Deviation from Cylindrical form t				End face circle runout t_1			
		shaft diameter		housing bore		shaft abutment		housing bore rib	
		bearing accuracy grade							
		0	6(6X)	0	6(6X)	0	6(6X)	0	6(6X)
over	to			tolerance value/ μm					
18	30	4.0	2.5	6	4.0	10	6	15	10
30	50	4.0	2.5	7	4.0	12	8	20	12
50	80	5.0	3.0	8	5.0	15	10	25	15
80	120	6.0	4.0	10	6.0	15	10	25	15
120	180	8.0	5.0	12	8.0	20	12	30	20
180	250	10.0	7.0	14	10.0	20	12	30	20
250	315	12.0	8.0	16	12.0	25	15	40	25
315	400	13.0	9.0	18	13.0	25	15	40	25
400	500	15.0	10.0	20	15.0	25	15	40	25

Table 3 - 38 Surface roughness of matching surfaces (unit: μm)

Shaft or housing bore diameter /mm		Shaft or housing matching surface diameter tolerance grade								
		IT7			IT6			IT5		
		surface roughness								
over	to	Rz	Ra		Rz	Ra		Rz	Ra	
			grinding	lathing		grinding	lathing		grinding	lathing
	80	10	1.6	3.2	6.3	0.8	1.6	4	0.4	0.8
80	500	16	1.6	3.2	10	1.6	3.2	6.3	0.8	1.6
end face		25	3.2	6.3	25	3.2	6.3	10	1.6	3.2

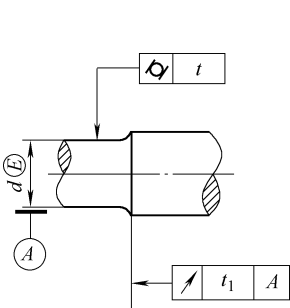


Figure 3 - 9 Marking method for geometric tolerance of shaft diameter

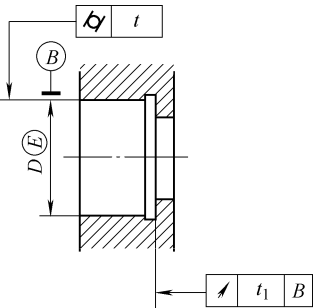


Figure 3 - 10 Marking method for geometric tolerance of housing bore

When using grade 2 bearings, Table 3 – 39 lists the geometric tolerance of the shaft and housing bore, Table 3 – 40 lists the geometric tolerance of the shaft abutment, housing bore rib and the washer, and Table 3 – 41 lists the surface roughness of matching surfaces. Figure 3 – 11, Figure 3 – 12 and Figure 3 – 13 are the marking methods for the geometric tolerand of the shaft, housing bore and washer.

Table 3 – 39 Geometric tolerance of shaft and housing bore
(matching to grade 2 bearing) (unit: μm)

Nominal diameter ^① /mm		Shaft diameter			Housing bore		
		deviation from circular form	deviation from cylindrical form	co- axis ^②	deviation from circular form	deviation from cylindrical form	co- axis ^②
		<i>t</i> ₁	<i>t</i> ₂	<i>t</i> ₃	<i>t</i> ₄	<i>t</i> ₅	<i>t</i> ₆
over ^③	to						
0.6	2.5	0.25	0.5	1.2	—	—	—
2.5	6	0.3	0.6	1.5	0.5	1	1.5
6	10	0.3	0.6	1.5	0.5	1	1.5
10	18	0.3	0.8	2	0.6	1.2	2
18	30	0.5	1	2.5	0.75	1.5	2.5
30	50	0.5	1	2.5	0.75	1.5	2.5
50	80	0.6	1.2	3	1	2	3
80	120	0.75	1.5	4	1.25	2.5	4
120	150	1	2	5	1.25	3.5	5
150	180	1	2	5	1.75	3.5	5
180	250	1.5	3	7	2.25	4.5	7
250	315	—	—	—	3	6	8
315	400	—	—	—	4.5	7	9

① See the Table 3 – 41 note.
② Designates the co-axis of two shaft diameters or two housing bores that apart from 300mm each other.
③ Include 0.6.

Table 3-40 Geometric tolerance of shaft abutment, housing bore rib and washer (matching to grade 2 bearing) (unit: μm)

Nominal diameter ^① /mm		Shaft abutment, housing bore rib		Washer	
		end face runout		end face runout ^②	two end face parallelism
over ^③	to	t_7		t_8	t_9
0.6	2.5	0.8	—	0.8	0.8
2.5	6	1	1	1	1
6	10	1	1	1	1
10	18	1.2	1.2	1.2	1.2
18	30	1.5	1.5	1.5	1.5
30	50	1.5	1.5	1.5	1.5
50	80	2	2	2	2
80	120	2.5	2.5	2.5	2.5
120	100	3.5	3.5	3.5	3.5
150	180	3.5	3.5	3.5	3.5
180	250	4.5	4.5	4.5	4.5
250	315	—	6	6	6
315	400	—	7	7	7

Note: Reference from CSBTS TC 98.75—1999.

① See the Table 3-41 note.

② Only suitable for the washer that is the extend part of the shaft abutment and the housing rib.

③ Include 0.6.

Table 3-41 Surface roughness (matching to grade 2 bearing)

Matching surface	Nominal diameter/mm		
	to 30	over 30 to 80	over 80 to 400
	R_a not larger/ μm		
shaft diameter	0.16	0.32	0.63
housing bore	0.32	0.63	1.25
shaft abutment bore rib washer end face	0.63	1.25	1.25

Note: 1. Nominal diameter designates the shaft or housing bore diameter, washer bore diameter. The surface roughness of the shaft diameter and shaft abutment end face can be checked out from relating table according to the diameter. The surface roughness of the housing bore surface and housing bore rib end face can be checked out according to the outer diameter.

2. Reference from CSBTS TC98.75—1999.

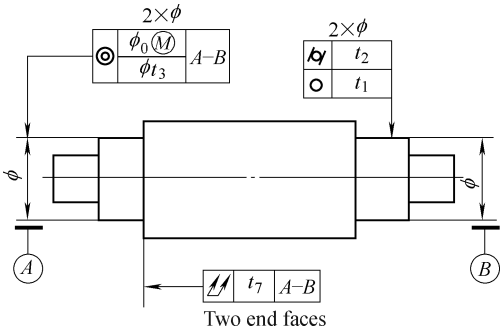


Figure 3-11 Marking method of the geometric tolerance of the shaft diameter matching to grade 2 bearing

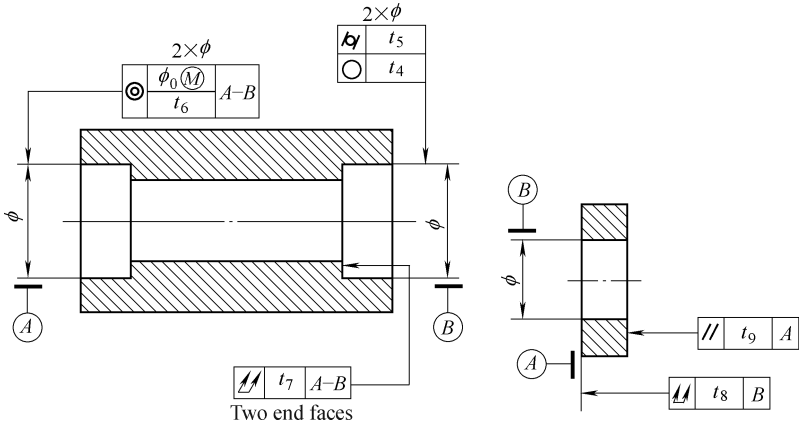


Figure 3-12 Marking method for the geometric tolerance of the housing bore matching to grade 2 bearing

Figure 3-13 Marking method for the geometric tolerance of the washer matching to grade 2 bearing

3.4 Rolling Bearing Floating Clearance Selection

3.4.1 Principles of Rolling Bearing Floating Clearance Selection

Rolling bearing floating clearance designates the arithmetic movement of the radial or axial directions through which one of the rings may be

displaced relative to the other, from one eccentric extreme position to the opposite extreme position without being subjected to any external load. For radial bearing, the theoretical original radial floating clearance is the outer ring raceway contact diameter minus the inner ring raceway contact diameter and minus twice the rolling element diameter.

The original radial floating clearance can be inspected for every bearing.

China National Standard GB/T4604—2006 stipulated the radial rolling bearing floating clearance as five groups: according to the clearance value it can be divided as clearance group 2, clearance group 0, clearance group 3, clearance group 4 and clearance group 5. The clearance group 2 is the minimum clearance value, the clearance group 5 is the maximum clearance value. The clearance group 0 is the basic clearance group having the priority for selection, which usually is not marked on the bearing symbol if selecting the clearance group 0.

The bearing working floating clearance is smaller than the original floating clearance because of the effects of interference fit and temperature. The bearing working floating clearance affects the bearing performance in many aspects, such as directly affecting the support running accuracy, load distribution in bearing interior, vibration and noise, friction torque, support rigidity, support self-aligning and so on. Therefore, it is reasonable to select the bearing floating clearance group according to the working condition and application performance. The basic radial bearing floating clearance group is suitable for the general running condition, common temperature, and commonly used interference fit, that is: for ball bearing, it is not over j5, k5 (shaft) and J6 (housing bore); for roller bearing it is not over k5, m5 (shaft) and K6 (housing bore). For the applications using tighter fit and the applications having the higher temperature difference between inner ring and outer ring and the applications of requiring to reduce the friction torque and the applications requiring to improve the self-aligning performance while the deep groove ball bearing subjecting to heavier axial load, it is recommended to select the larger floating clearance group. For

the applications requiring higher running accuracy or the applications limiting the axial movement, it is suitable to select the smaller floating clearance group.

For ball bearing, the most suitable working floating clearance tends to be zero. For roller bearing, it is better to keep a small working floating clearance. For requiring good support rigidity (such as the spindle of machine tool) there should be some preload.

Angular contact ball bearings, tapered roller bearings and bearings with tapered bore of inner ring, because of their structure design, the bearing floating clearance is obtained through the adjustment during installation.

3.4.2 Selecting Rolling Bearing Clearance by Estimation Calculation

It is very difficult to calculate precisely the required original bearing floating clearance. But the required clearance group can be selected by estimating the reduced amount of radial floating clearance caused by the interference fit.

There is a reference that introduces an estimation method for calculating the reduced amount of radial floating clearance caused by the interference fit that is suitable for solid shaft, hollow shaft, thick wall housing, and thin wall housing.

The bearing original radial floating clearance will be reduced by the inner ring raceway expansion and the outer ring raceway shrinkage. The radial floating clearance reduced amount is related to the actual effective interference amount of matching parts, the matching shaft diameter dimension and the wall thickness of the housing. The actual effective interference amount Δd_y and ΔD_y designate the interference amount that the nominal interference amount Δd and ΔD are reduced because the rough and uneven matching surfaces are flattened after the matching parts with interference fit are pressed into each other. The more rough the contacting surface is more, the more the flattened amount is. But bearing inner ring and outer ring both are manufactured by quenching followed by precision

grinding, therefore, the flattened amount is very little that can be omit. The flattened amount G^* of the shaft diameter surface or the housing bore surface matching with bearing can be selected from Table 3 – 42 according to the surface final manufacturing method. During calculating the effective interference amount, it should also consider the value range of nominal interference amount. Usually, in order to avoid the waste product in manufacturing process, the product dimensions are controlled to be the upper end of the tolerance band. Therefore, when takes the mominal interference amount value, the original tolerance band of matching part should be reduced 1/3 from the upper end, so, the actual effective interference amount should be as:

$$\Delta d_y = 2\Delta d/3 - G^*$$

Δd —nominal interference amount, such as:

bearing bore diameter $\phi 40$	tolerance ${}_{-12}^0 \mu\text{m}$
shaft $\phi 40 \text{ k5}$	tolerance ${}_{+2}^{+13} \mu\text{m}$
after shrinking 1/3	
bearing bore diameter $\phi 40$	tolerance ${}_{-8}^0 \mu\text{m}$
shaft $\phi 40 \text{ k5}$	tolerance ${}_{+2}^{+9} \mu\text{m}$
actual effective interference amount	$+17 \mu\text{m}$

Table 3 – 42 The flattened amount of interference fit G^* (unit: μm)

Matching surface final manufacturing	Matching component roughness R_t	Diameter flattened amount G^*
precion grinding	0. 8	≈ 1
fine grinding	2	$\approx 2. 5$
precision lathing	4	≈ 5
fine lathing	6	≈ 7

Figure 3 – 14 is the gemetric dimension after matching bearing inner ring to shaft.

Figure 3 – 15 is the gemetric dimension after matching bearing outer ring to housing bore.

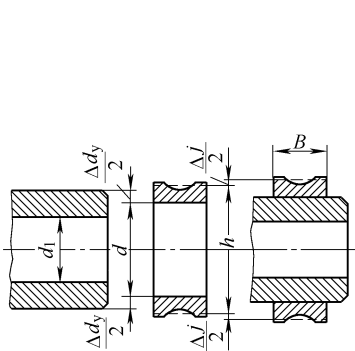


Figure 3 - 14 Geometric dimension after matching bearing inner ring to shaft

Note: The symbols in figure see Table 3 - 43

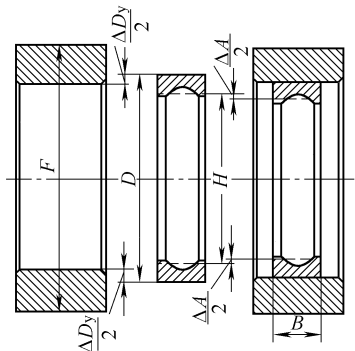


Figure 3 - 15 Geometric dimension after matching outer ring to housing bore

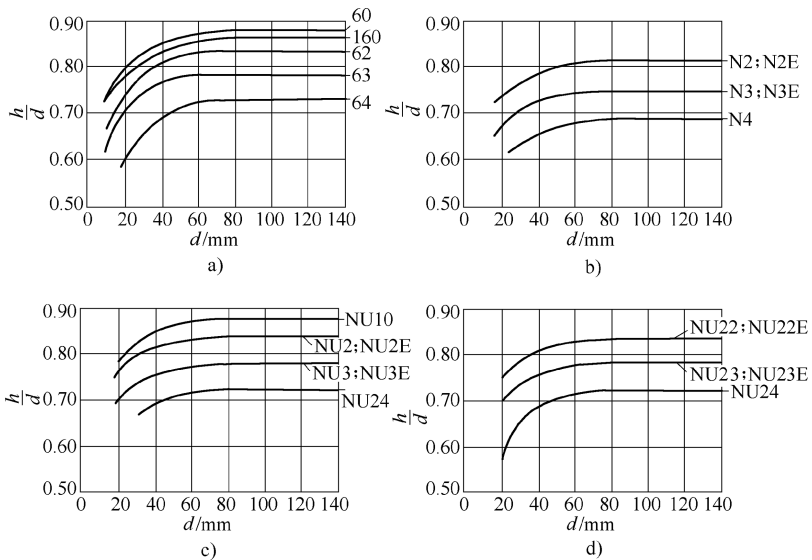
Note: The symbols in figure see Table 3 - 43

Table 3 - 43 Estimating equations for calculating radial floating clearance reduced amount

Matching component	$\Delta j, \Delta A$ calculating equation	symbol
bearing inner ring to steel solid shaft	$\Delta j = \Delta d_y \cdot d/h$ (1)	Δj —inner ring raceway rib diameter expansion amount (μm)
bearing inner ring to steel hollow shaft	$\Delta j = \Delta d_y \cdot F(d)$ (2) $F(d) = d/h \frac{(d/d_1)^2 - 1}{(d/d_1)^2 - (d/h)^2}$ (2a)	Δd_y —shaft diameter effective interference amount (μm) d —bearing inner ring diameter (mm) h —inner ring raceway rib diameter (mm)
bearing outer ring to steel solid housing	$\Delta A = \Delta D_y \cdot H/D$ (3)	B —bearing width (mm) d_1 —hollow shaft inner diameter (mm)
bearing outer ring to steel thin wall housing	$\Delta A = \Delta D_y \cdot F(D)$ (4) $F(D) = H/D \frac{(F/D)^2 - 1}{(F/D)^2 - (H/D)^2}$ (4a)	ΔA —outer ring raceway rib diameter reduced amount (μm) ΔD_y —housing bore diameter actual effective interference amount (μm)
bearing outer ring to cast iron housing	$\Delta A = \Delta D_y \cdot [F(D) - 0.15]$ (5)	H —outer ring rib diameter (mm)
bearing outer ring to light metal housing	$\Delta A = \Delta D_y \cdot [F(D) - 0.25]$ (6)	D —bearing outer ring and housing bore diameter (mm) F —bearing housing outer diameter (mm)

In order to calculate easily, d/h and H/D in the equations (1), (3) in Table 3-43 can be expressed by the curve lines of $d-d/h$, $D-H/D$. When the bearing type and model has been confirmed, the bearing structure dimensions become known dimensions, such as the bearing bore diameter d , the inner ring rib diameter h , the outer ring diameter D and the outer ring rib diameter H . Figure 3-16 is the commonly used bearing $d-d/h$ curve group. While calculating the inner ring raceway expansion amount Δj , first we check out d/h from d , then put the checking value into equation (1) in Table 3-43 to get the Δj .

Figure 3-17 is the commonly used bearing $D-H/D$ curve group. While calculating the outer ring raceway shrinking amount ΔA , first we check out H/D from D , then put the checking value into equation (3) in Table 3-43 to get the ΔA .



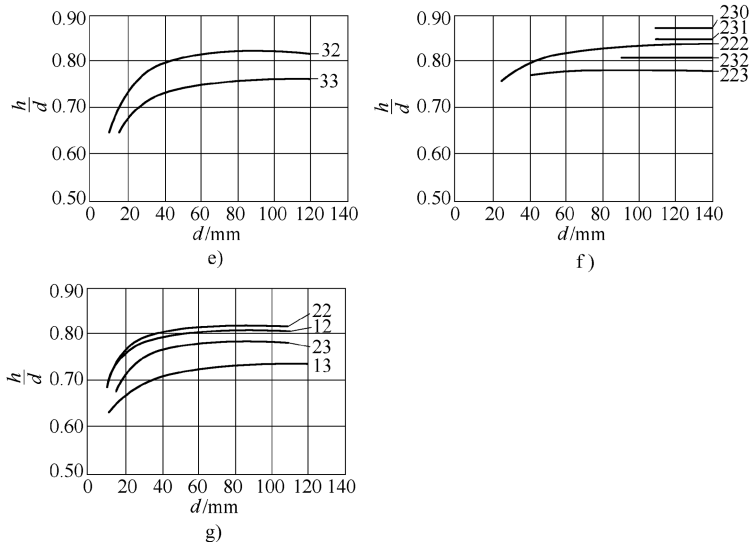
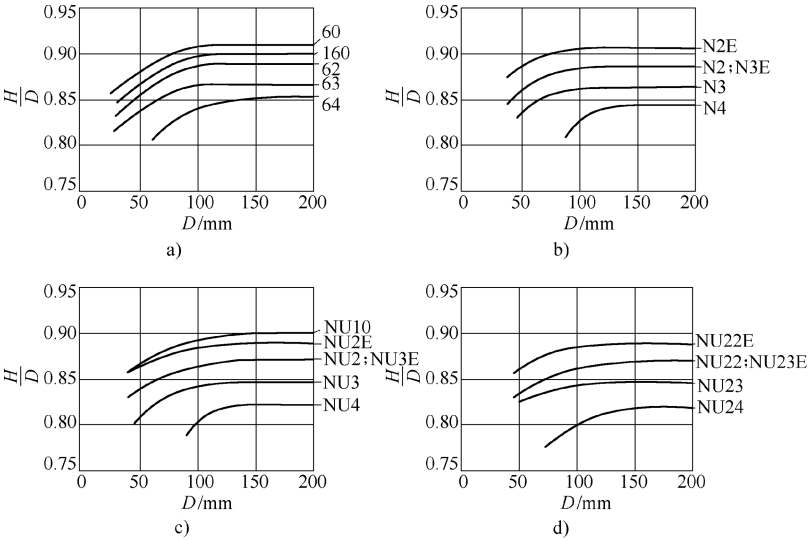


Figure 3 - 16 Inner ring d — d/h curves

- a) deep groove ball bearing b) cylindrical roller bearing with inner ring with rib
c) cylindrical roller bearing with outer ring with rib d) cylindrical roller bearing with outer ring with rib
e) double-row angular contact ball bearing f) self-aligning roller bearing
g) self-aligning ball bearing



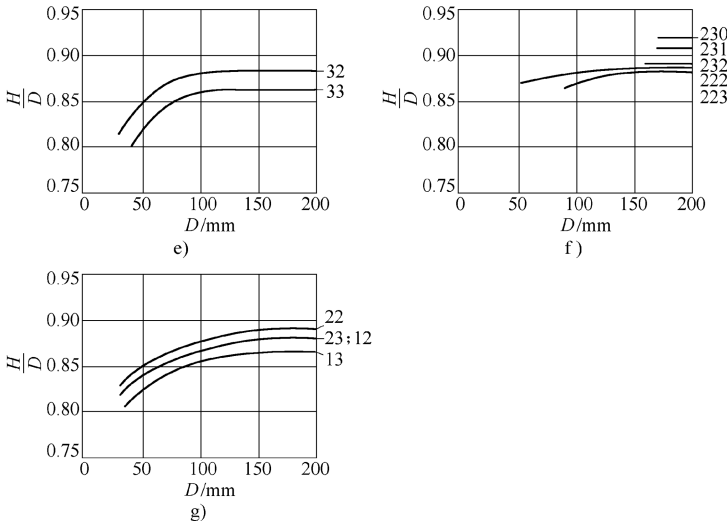


Figure 3-17 Outer ring D — H/D curves

- a) deep groove ball bearing b) cylindrical roller bearing with inner ring with rib
 c) cylindrical roller bearing with outer ring with rib d) cylindrical roller bearing with outer ring with rib e) double-row angular contact ball bearing f) self-aligning roller bearing
 g) self-aligning ball bearing

In order to simplify the calculation of Δj and ΔA of hollow shaft and thin wall housing, we simplify the functions $F(d)$ and $F(D)$ in the equations (2a), (4a) in Table 3-43, taking the d/d_1 , F/D as constant values to get the d/h — $\Delta j/\Delta d_y$ and H/D — $\Delta A/\Delta D_y$ curves. Because the d/d_1 , F/D can take different

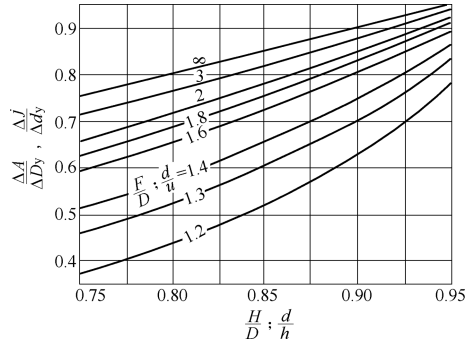


Figure 3-18 Hollow shaft and thin wall housing Δj , ΔA calculating curves

values, therefore, Figure 3-18 is a group of curves.

Calculating procedure of hollow shaft inner ring raceway expansion

amount is:

$$d \rightarrow d/h, \quad d/h \rightarrow \Delta j / \Delta d \quad (d/d_1 = \text{constant}), \quad \Delta j = (d/h) \Delta d_y$$

Calculating procedure of thin wall housing outer ring raceway shrinking amount is:

$$D \rightarrow H/D, \quad H/D \rightarrow \Delta A / \Delta D_y \quad (F/D = \text{constant}), \quad \Delta A = (H/D) \Delta D_y$$

When the housing is made by cast iron or light metal material, the outer ring raceway shrinking amount ΔA calculating procedure is the same as the thin wall housing. But because of the modulus of elasticity is different, the calculating result of the ΔA must be amended:

For cast iron housing, $\Delta A / \Delta D_y$ should be reduced for 0.15, so

$$\Delta A = \Delta D_y \quad [F(D) - 0.15]$$

For light metal housing, $\Delta A / \Delta D_y$ should be reduced for 0.25, so

$$\Delta A = \Delta D_y \quad [F(D) - 0.25]$$

[**Example 3 – 2**] Figure 3 – 19 is a support structure of bridge crane roller. This support has two self-aligning roller bearings floatingly mounted on the same separating sleeve. There has been left a clearance of 0.5 ~ 1mm between the bearing inner ring and the separating sleeve rib for compensating the mounting error and the sleeve axial direction floating.

Working data:

span distance	24 m
car weight	175kN
crane big beam weight	920kN
hoist weight	245kN
moving speed	2 m/s
location parameter when car lying on end	1.8 m
roller diameter	900mm
roller width	400mm
bearing span distance	250mm
bearing model	22332
bearing original data	see Table 3 – 44

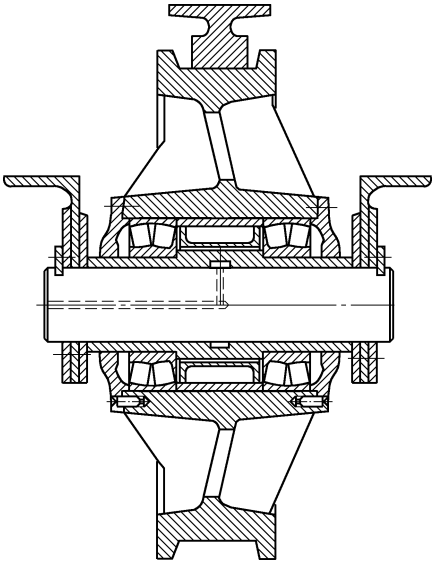


Figure 3 - 19 Track rollersupport

Table 3 - 44 Bearing original data

bearing speed / (r/min)	support load /kN	rating load /kN	life factor f_L	shaft manufacture $d = 160\text{mm}$	housing manufacture $D = 340\text{mm}$	housing outer diameter /mm
42	334	920	3. 6	procise griding	procise lathing	440

Note: Inner ring, subjecting to local heavy load , floating in axial direction relative to the separate sleeve, taking g6 fit.

Outer ring, subjecting to circulating load, tight fit to the housing, taking P6 fit.

Because there is no other heat entering and the temperature difference of inner ring and outer ring is not too large, therefore, we only need to calculate the radial floating clearance reduced amount Δj , ΔA due to interference amount. (calculating results are listed in Table 3 - 45).

The radial floating clearance reduced amount $\Delta j + \Delta A = 4\mu\text{m} + 31\mu\text{m} \approx 35\mu\text{m}$

According to the clearance range, select the clearance group 2 (the radial floating clearance of clearance group 2 is $60 \sim 110\mu\text{m}$).

Table 3 – 45 Radial clearance reduced amount from interference amount

Calculation item	Inner ring and separate sleeve	Outer ring and housing bore	Illustration
maximum nominal interference amount $\Delta d, \Delta D/\mu\text{m}$	11	98	based on Table 3 – 25 , Table 3 – 26
flattened amount $G^*/\mu\text{m}$	2. 5	7	based on Table 3 – 42
maximum effective in- terference amount $\Delta d_y,$ $\Delta D_y/\mu\text{m}$	4. 8	58	based on equation $\Delta d_y = (2/3) \Delta d - G^*$ $\Delta D_y = (2/3) \Delta D - G^*$
$d/h, D/H$	0. 78	0. 885	based on Figure 3 – 16 , Figure 3 – 17
$\Delta j/\mu\text{m}$	≈ 4		based on Figure 3 – 18
$\Delta A/\mu\text{m}$		≈ 31	$F/D = 1. 3$ $\Delta A/\Delta D_y = F(D) = 0. 69$ $\Delta A = \Delta D_y [F(D) - 0. 15]$ $= 58 [0. 69 - 0. 15] \approx 31$

3. 4. 3 Radial Floating Clearance of Every Type of Rolling Bearings

Radial floating clearance of every type radial bearing is listed in Table 3 – 46 to Table 3 – 53.

Table 3 – 46 Deep groove ball bearing radial floating clearance

(unit: μm)

d/mm		group 2		group 0		group 3		group 4		group 5	
over	to	min	max	min	max	min	max	min	max	min	max
2. 5	6	0	7	2	13	8	23	—	—	—	—
6	10	0	7	2	13	8	23	14	29	20	37
10	18	0	9	3	18	11	25	18	33	25	45
18	24	0	10	5	20	13	28	20	36	28	48
24	30	1	11	5	20	13	23	23	41	30	53
30	40	1	11	6	20	15	33	28	46	40	64
40	50	1	11	6	23	18	36	30	51	45	73

(continued)

<i>d</i> /mm		group 2		group 0		group 3		group 4		group 5	
over	to	min	max	min	max	min	max	min	max	min	max
50	65	1	15	8	28	23	43	38	61	55	90
65	80	1	15	10	30	25	51	46	71	65	105
80	100	1	18	12	36	30	58	53	84	75	120
100	120	2	20	15	41	36	66	61	97	90	140
120	140	2	23	18	48	41	81	71	114	105	160
140	160	2	23	18	53	46	91	81	130	120	180
160	180	2	25	20	61	53	102	91	147	135	200
180	200	2	30	25	71	63	117	107	163	150	230
200	225	2	35	25	85	75	140	125	195	175	265
225	250	2	40	30	95	85	160	145	225	205	300
250	280	2	45	35	105	90	170	155	245	225	340
280	315	2	55	40	115	100	190	175	270	245	370
315	355	3	60	45	125	110	210	195	300	275	410
355	400	3	70	55	145	130	240	225	340	315	460
400	450	3	80	60	170	150	270	250	380	350	510
450	500	3	90	70	190	170	300	280	420	390	570
500	560	10	100	80	210	190	330	310	470	440	630
560	630	10	110	90	230	210	360	340	520	490	690
630	710	20	130	110	260	240	400	380	570	540	760
710	800	20	140	120	290	270	450	430	630	600	840
800	900	20	160	140	320	300	500	480	700	670	940
900	1000	20	170	150	350	330	550	530	770	740	1040
1000	1120	20	180	160	380	360	600	580	850	820	1150
1120	1250	20	190	170	410	390	650	630	920	890	1260

Table 3 – 47 Cylindrical bore self-aligning ball bearing radial floating clearance

(unit: μm)

<i>d</i> /mm		group 2		group 0		group 3		group 4		group 5	
over	to	min	max	min	max	min	max	min	max	min	max
2.5	6	1	8	5	15	10	20	15	25	21	33
6	10	2	9	6	17	12	25	19	33	27	42
10	14	2	10	6	19	13	26	21	35	30	48
14	18	3	12	8	21	15	28	23	37	32	50
18	24	4	14	10	23	17	30	25	39	34	52
24	30	5	16	11	24	19	35	29	46	40	58
30	40	6	18	13	29	23	40	34	53	46	66
40	50	6	19	14	31	25	44	37	57	50	71
50	65	7	21	16	36	30	50	45	69	62	88
65	80	8	24	18	40	35	60	54	83	76	108
80	100	9	27	22	48	42	70	64	96	89	124
100	120	10	31	25	56	50	83	75	114	105	145
120	140	10	38	30	68	60	100	90	135	125	175
140	160	15	44	35	80	70	120	110	161	150	210

Table 3 – 48 Tapered bore self-aligning ball bearing radial floating clearance

(unit: μm)

<i>d</i> /mm		group 2		group 0		group 3		group 4		group 5	
over	to	min	max	min	max	min	max	min	max	min	max
18	24	7	17	13	26	20	33	28	42	37	55
24	30	9	20	15	28	23	39	33	50	44	62
30	40	12	24	19	35	29	46	40	59	52	72
40	50	14	27	22	39	33	52	45	65	58	79
50	65	18	32	27	47	41	61	56	80	73	99
65	80	23	39	35	57	50	75	69	98	91	123
80	100	29	47	42	68	62	90	84	116	109	144
100	120	35	56	50	81	75	108	100	139	130	170
120	140	40	68	60	98	90	130	120	165	155	205
140	160	45	74	65	110	100	150	140	191	180	240

Table 3-49 Radial floating clearance of cylindrical bore of cylindrical roller bearing(unit: μm)

d/mm		group 2		group 0		group 3		group 4		group 5	
over	to	min	max	min	max	min	max	min	max	min	max
	10	0	25	20	45	35	60	50	75	—	—
10	24	0	25	20	45	35	60	50	75	65	90
24	30	0	25	20	45	35	60	50	75	70	95
30	40	5	30	25	50	45	70	60	85	80	105
40	50	5	35	30	60	50	80	70	100	95	125
50	65	10	40	40	70	60	90	80	110	110	140
65	80	10	45	40	75	65	100	90	125	130	165
80	100	15	50	50	85	75	110	105	140	155	190
100	120	15	55	50	90	85	125	125	165	180	220
120	140	15	60	60	105	100	145	145	190	200	245
140	160	20	70	70	120	115	165	165	215	225	275
160	180	25	75	75	125	120	170	170	220	250	300
180	200	35	90	90	145	140	195	195	250	275	330
200	225	45	105	105	165	160	220	220	280	305	365
225	250	45	110	110	175	170	235	235	300	330	395
250	280	55	125	125	195	190	260	260	330	370	440
280	315	55	130	130	205	200	275	275	350	410	485
315	355	65	145	145	225	225	305	305	385	455	535
355	400	100	190	190	280	280	370	370	460	510	600
400	450	110	210	210	310	310	410	410	510	565	665
450	500	110	220	220	330	330	440	440	550	625	735

Note: Needle roller bearing radial clearance:

Needle roller bearing with inner ring, outer ring and cage can use this Table value for cylindrical roller bearing radial clearance, except the draw cup needle roller bearing and heavy series needle roller bearing.

Needle roller bearing with inner ring, outer ring of heavy series and inner ring as a separate component supplying, its radial clearance should be decided by the inner ring raceway diameter and the needle roller set inner ring diameter. These diameter tolerance stipulated in China GB/T 5801—2006.

**Table 3 – 50 Radial floating clearance of cylindrical
bore of self-aligning roller bearing** (unit: μm)

<i>d</i> /mm		group 2		group 0		group 3		group 4		group 5	
over	to	min	max	min	max	min	max	min	max	min	max
14	18	10	20	20	35	35	45	45	60	60	75
18	24	10	20	20	35	35	45	45	60	60	75
24	30	15	25	25	40	40	55	55	75	75	95
30	40	15	30	30	45	45	60	60	80	80	100
40	50	20	35	35	55	55	75	75	100	100	125
50	65	20	40	40	65	65	90	90	120	120	150
65	80	30	50	50	80	80	110	110	145	145	180
80	100	35	60	60	100	100	135	135	180	180	225
100	120	40	75	75	120	120	160	160	210	210	260
120	140	50	95	95	145	145	190	190	240	240	300
140	160	60	110	110	170	170	220	220	280	280	350
160	180	65	120	120	180	180	240	240	310	310	390
180	200	70	130	130	200	200	260	260	340	340	430
200	225	80	140	140	220	220	290	290	380	380	470
225	250	90	150	150	240	240	320	320	420	420	520
250	280	100	170	170	260	260	350	350	460	460	570
280	315	110	190	190	280	280	370	370	500	500	630
315	355	120	200	200	310	310	410	410	550	550	690
355	400	130	220	220	340	340	450	450	600	600	750
400	450	140	240	240	370	370	500	500	660	660	820
450	500	140	260	260	410	410	550	550	720	720	900
500	560	150	280	280	440	440	600	600	780	780	1000
560	630	170	310	310	480	480	650	650	850	850	1100
630	710	190	350	350	530	530	700	700	920	920	1190
710	800	210	390	390	580	580	770	770	1010	1010	1300
800	900	230	430	430	650	650	860	860	1120	1120	1440
900	1000	260	480	480	710	710	930	930	1220	1220	1570

**Table 3-51 Radial floating clearance of tapered
bore of self-aligning roller bearing** (unit: μm)

d/mm		group 2		group 0		group 3		group 4		group 5	
over	to	min	max	min	max	min	max	min	max	min	max
18	24	15	25	25	35	35	45	45	60	60	75
24	30	20	30	30	40	40	55	55	75	75	95
30	40	25	35	35	50	50	65	65	85	85	105
40	50	30	45	45	60	60	80	80	100	100	130
50	65	40	55	55	75	75	95	95	120	120	160
65	80	50	70	70	95	95	120	120	150	150	200
80	100	55	80	80	110	110	140	140	180	180	230
100	120	65	100	100	135	135	170	170	220	220	280
120	140	80	120	120	160	160	200	200	260	260	330
140	160	90	130	130	180	180	230	230	300	300	380
160	180	100	140	140	200	200	260	260	340	340	430
180	200	110	160	160	220	220	290	290	370	370	470
200	225	120	180	180	250	250	320	320	410	410	520
225	250	140	200	200	270	270	350	350	450	450	570
250	280	150	220	220	300	300	390	390	490	490	620
280	315	170	240	240	330	330	430	430	540	540	680
315	355	190	270	270	360	360	470	470	590	590	740
355	400	210	300	300	400	400	520	520	650	650	820
400	450	230	330	330	440	440	570	570	720	720	910
450	500	260	370	370	490	490	630	630	790	790	1000
500	560	290	410	410	540	540	680	680	870	870	1100
560	630	320	460	460	600	600	760	760	980	980	1230
630	710	350	510	510	670	670	850	850	1090	1090	1360
710	800	390	570	570	750	750	960	960	1220	1220	1500
800	900	440	640	640	840	840	1070	1070	1370	1370	1690
900	1000	490	710	710	930	930	1190	1190	1520	1520	1860

**Table 3 – 52 Recommended radial clearance of tapered
bore of double-row cylindrical roller bearing (unit: μm)**

<i>d</i> /mm		group 1 ^①		group 2	
over	to	min	max	min	max
	24	10	20	20	30
24	30	15	25	25	35
30	40	15	25	25	40
40	50	17	30	30	45
50	65	20	35	35	50
65	80	25	40	40	60
80	100	35	55	45	70
100	120	40	60	50	80
120	140	45	70	60	90
140	160	50	75	65	100
160	180	55	85	75	110
180	200	60	90	80	120
200	225	60	95	90	135
225	250	65	100	100	150
250	280	75	110	110	165
280	315	80	120	120	180
315	355	90	135	135	200
355	400	100	150	150	225
400	450	110	170	170	255
450	500	120	190	190	285

① The clearance value of clearance group 1 is smaller than the clearance value of clearance group 2. Same in next Table.

**Table 3 – 53 Recommended radial clearance of cylindrical
bore of double-row cylindrical roller bearing (unit: μm)**

<i>d</i> /mm		group 1		group 2		group 3	
over	to	min	max	min	max	min	max
	24	5	15	10	20	20	30
24	30	5	15	10	25	25	35
30	40	5	15	12	25	25	40
40	50	5	18	15	30	30	45
50	65	5	20	15	35	35	50

(continued)

d/mm		group 1		group 2		group 3	
over	to	min	max	min	max	min	max
65	80	10	25	20	40	40	60
80	100	10	30	25	45	45	70
100	120	10	30	25	50	50	80
120	140	10	35	30	60	60	90
140	160	10	35	35	65	65	100
160	180	10	40	35	75	75	110
180	200	15	45	40	80	80	120
200	225	15	50	45	90	90	135
225	250	15	50	50	100	100	150
250	280	20	55	55	110	110	165
280	315	20	60	60	120	120	180
315	355	20	65	65	135	135	200
355	400	25	75	75	150	150	225
400	450	25	85	85	170	170	255
450	500	25	95	95	190	190	285

3.5 Rolling Bearing Sealing Device

Seals are indispensable for bearings. Seals have dual purposes of retaining the lubricant within the bearing and preventing the ingress of contaminants. Otherwise, abrasive particles not only can cause raceway wear, shorten the service life, but also can cause bearing parts rust by harmful gas and water, speeding the lubricant ageing. Therefore, bearing seal device design is an important content of bearing system. In designing, it should be considered to keep long time sealing function and to prevent the ingress of contaminants, and at the same time, to have little friction and mounting error, easy mounting, dismounting and simple maintainance and care.

Generally, sealing device can be divided as two types: stationary seals (fixed seals) and moving seals (rotating seals). The former is called as washer seals, and the latter is called as seal ring seals. According to the seal structure, sealing device also can be divided as: contact seal and non-contact seal.

The contact seal is that there is certain closing pressure between the

sealing device and the required sealing position for their directly contacting. Therefore, there are so many factors that would directly influence the bearing friction torque, the permissible rotating speed and the temperature rise, such as the contact pattern, the closing pressure, the lubrication condition, the sliding speed and the manufacturing quality of the contacting surfaces. Table 3 – 54 lists the permissible circumference speed, Table 3 – 55 lists the requirements for the surfaces of closing contacting. For all of contact seal device, there will be wear during bearing running. The wear degree and failure level are related to the contact seal self-performance and the application conditions.

The non-contact seal is that there is no directly contacting between the sealing device and the required sealing position. Because there is no other friction existing except lubricant friction in between the sealing gap, there is no wear produced and no obvious heat generated after long time usage in the non-contact seal, therefore, this non-contact seal is suitable for higher rotating speed applications. But the sealing gap can not be too large, otherwise, the sealing effect would be lost. Table 3 – 56 lists the gap clearance of the non-contact sealing device.

Table 3 – 54 Permissible circumference speed of contact seals

Material	Permissible circumference speed/ (m/s)
felt	3.5 ~ 4
leather	8 ~ 10
synthetic rubber	10 ~ 15

Table 3 – 55 Close contact surface requirement

Circumference speed/ (m/s)	Surface roughness $Ra/\mu\text{m}$	Manufacturing requirement
~ 5	3.2	grinding
5 ~ 10	1.6 ~ 3.2	grinding
over 10	0.8	hardening, chromium plating

Table 3 – 56 Non-contact sealing device gap clearance (unit: mm)

Shaft diameter	Radial clearance	Axial clearance
< 5	0.20 ~ 0.40	1 ~ 2
≥ 50	0.50 ~ 1.00	3 ~ 5

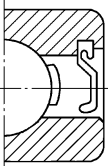
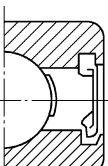
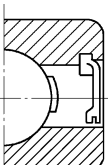
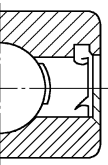
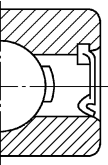
The bearing sealing device can not only install on the support location, but also can install on bearing itself. The former is called support seal, and the latter is called self-seal.

3.5.1 Bearing Self-Sealing

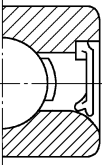
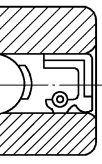
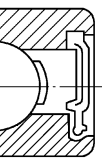
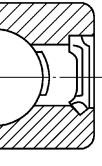
The sealing device of bearing support can effectively prevent the ingress of contaminants, but the additional sealing device will increase the machinery axial dimension, so that the bearing usage, installation and maintenance are inconvenient, at the same time, the manufacturing accuracy and geometric tolerance of the matching surfaces of the shaft and the housing also can affect its sealing function. The bearing with seals has the advantages of having a good sealing and lubrication surroundings as well as resolving the problems of the support sealing, at the same time, a complicated lubrication system can be saved. Therefore, it is recommended for all applications of requiring compact structure and strict sealing to use the bearing with seals. Besides the bearing with seals have very good effect of dustproof and sealing, it could keep continuous running without supplying lubricant during the running process and reducing the maintenance cost.

According to the bearing self-sealing structures, they can be divided as contact sealing and non-contact sealing. According to the contact forms, they can be divided as radial sealing and axial sealing. According to the sealing ring structures, they can be divided as single-lip sealing and double-lip sealing. Generally, there are two types of non-contact sealings: with shield and with rubber seals. The shield sealing clearance is larger than the rubber sealing clearance, so the bearings with shield are only suitable for dust proof. The bearings with rubber seals have good sealing effect because of their smaller clearance. Generally, contact sealing bearing only has the rubber seals and its auxiliary parts. The rubber seals lip edge contacts the bearing rotating location in interference contact with certain closing force, thus its sealing effect is very good, but the rotating speed limit is lower and the friction torque is larger. Table 3 - 57 lists the sealing types and characteristics. Table 3 - 58 lists the commonly used sealing rubber property.

Table 3 –57 Sealing types and characteristics

Form	Sketch	Feature	Characteristics				
			dust proof	high speed	leaking proof	sealing space	dynamic effect
non- contact	shiled  a) JZ form	non-contact steel plate shield (commonly used)	*	⊙	*	○	—
	singlelip seals  b) RU form	steel plate and rubber seals, having groove on inner ring forming laby- rinth sealing	△	⊙	△	○	—
	doublelip seals  c)	steel plate and rubber non-contact seals (most commonly used)	△	⊙	○	△	—
contact	radial sealing  d)	steel plate and rubber contact seals (most commonly used)	○	△	○	⊙	△
	 e) RS form	steel plate and rubber contact seals (most commonly used)	○	△	○	○	△

(continued)

Form		Sketch	Feature	Characteristics				
				dust proof	high speed	leaking proof	sealing space	dynamic effect
contact	radial sealing	 f) RK form	steel plate and rubber contact seals, same as RS form, but one more lip edge	⊙	△	⊙	○	○
	radial sealing	 g) HMS(A) MNS(A) form	steel plate and rubber contact seals, sealing lip by preload force on inner ring	⊙	△	⊙	△	⊙
	axial sealing	 h) RD form	steel plate and rubber contact seals, double-lip structure, one lip contacts inner ring abutment, another contacts inner ring face forming labyrinth	○	○	○	○	○
		 i)	steel plate and rubber contact seals, patent	○	○	○	○	⊙

Note: ⊙— excellent ○— good △— general *— bad

Table 3 – 58 Commonly used sealing rubber property

Rubber material		NBR	AER (PA)	SR	FR	CR	SBR	11 R
mechanics	hardness	70 ~ 80	70 ~ 80	80	80	60 ~ 80	60 ~ 80	50 ~ 80
	deformation	◎	○	◎	○	◎	◎	○
	no-lubricant	◎	○	*	○	○	○	○
	wearproof	◎	○	*	○	◎	◎	○
	endurance	◎	○	*	◎	◎	◎	◎
	anti-tearing	◎	○	*	◎	◎	◎	◎
heat resistance	lowest temperature/°C	- 50	- 20	- 60	- 40	- 50	- 50	- 60
	max. temperature/°C	135	170	250	220	120	120	130

Advantages: NBR—cheap, good oil proof and wear proof, good anti-cold resistance and good soaking expanding proof, comprehensive usage range

AER (PA)—little reacting for ultra-pressure additive, good heat resistance, good soaking expanding proof

SR—good heat resistance, good low temperature resistance

FR—suitable for most of additives, good heat resistance, good soaking expanding proof

CR—good fitting to temperature, wear proof, good recovery elasticity

SBR—little reacting to ethyl and water alcohol acid

11R—ozone resistance, good ageing resistance, little reacting to non-burning hydraulic oil

Disadvantages: NBR—bad heat resistance

AER (PA)—bad low temperature resistance

SR—worst soaking expanding proof, not suitable for acid oil and ultra-pressure additive

FR—high cost, need special metal model for manufacturing

CR—having soaking expanding for halide, fragrant compound CS₂ phenol

SBR—having soaking expanding for gasoline

11R—having soaking expanding for mineral oil, having solubility for gasoline

note: ◎—excellent ○— good * — bad

3. 5. 2 Support Sealing

The bearing support sealing designates the additional sealing device

located outside the bearing, such as the sealing device on the housing position, shaft diameter position and the end cap position. Table 3 – 59 lists the commonly used bearing support sealing devices.

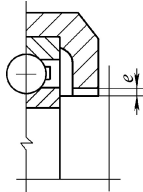
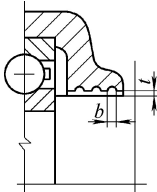
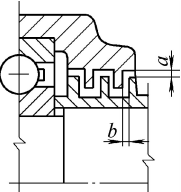
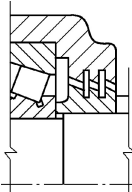
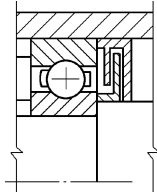
When selecting bearing support sealing device, many factors should be taken into account, such as the bearing lubricant type (grease or oil), the sealing contacting surface circumference speed, the mounting error and so on. For the applications that the sealing requirement is not very serious, but the bearing rotating speed is high and the mounting error is large, it is recommended to select the non-contact sealing device that can be divided as two types: sealing by the clearance and sealing by the centrifugal force of the centrifugal plate. In order to increase the sealing effect, it is recommended to add additional centrifugal plate or obstacle plate for getting the sealing effect by centrifugal force and controlling the grease draining amount. If the working surrounding is in hostile condition that there is the possibility of water invading, and if there is the possibility of using oil lubrication, it is recommended to select the contact sealing that can be divided as oil sealing, grease sealing, and machinery sealing.

3. 5. 3 Sealing Purpose and Structure Treatment

All kinds of sealing component could be used in a certain working condition and surrounding condition separately according to its working principle, but are always installed to finish all sealing tasks on one mounting position, that is to achieve all sealing purposes by the above sealing device. Therefore, the sealing device must suit to the entire structure. The usage life, working reliability and simple structure are the main aspects for consideration.

Table 3 – 60 lists the most of support surrounding conditions and its sealing devices. Through these examples we could get the overall view about the sealing purpose and the concrete treatment way. These are not only suitable for the listed examples, but also suitable for the resembling surrounding conditions and other supports.

Table 3 - 59 Bearing support sealing device

Non-contact	Sketch					
	type	narrow gap	groove sealing	labyrinth sealing	labyrinth with inclined paths	labyrinth with steel plate
	feature	simple, can meet general sealing asking clearance (radius) $d \leq 50 \text{ mm}$ $e = 0.25 \sim 0.40 \text{ mm}$ $d > 50 \text{ mm}$ $e = 0.25 \sim 0.6 \text{ mm}$	grease in groove preventing the ingress of contaminants; grooves also can drain grease. forming spiral; can be 3 grooves. groove width $b: 3 \sim 5 \text{ mm}$, depth $t: 4 \sim 5 \text{ mm}$	good sealing effect, radial and axial clearance: lower than 50 mm $a = 0.25 \sim 0.40 \text{ mm}$ $b = 1 \sim 2 \text{ mm}$ under 50 ~ 200 mm $a = 0.5 \sim 1.5 \text{ mm}$ $b = 2 \sim 5 \text{ mm}$	using in large deflection. can swing filling grease, having good sealing effect.	pressed plate making Labyrinth sealing, need no axial fix, simple, good seal result

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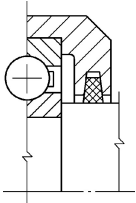
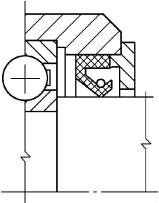
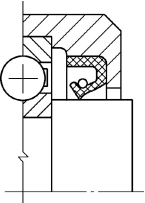
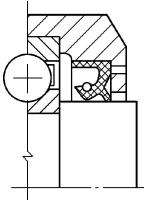
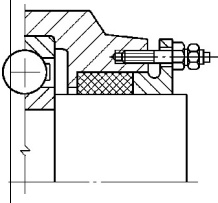
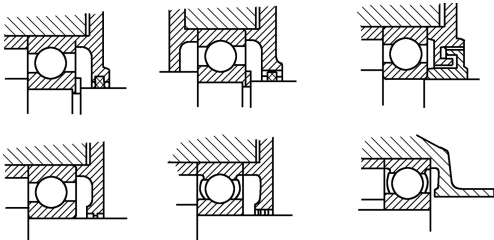
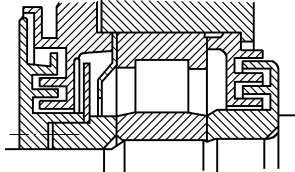
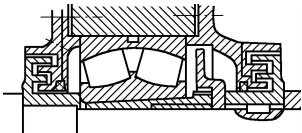
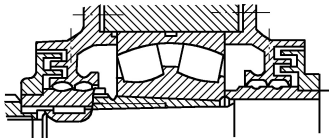
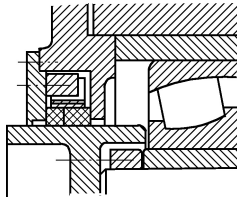
Contact	Sketch					
	type	felt sealing	outside lip sealing	inside lip sealing	double lip sealing	filling sealing
	feature	using under 100℃ , felt soaked before using, good sealing effect ,circ- umference speed lower than 4m/s	mainly preventing lu- bricant overflow, speed decided by sealing mate- rial	can prevent contami- nants invading, speed decided by sealing mate- rial	prevent lubricant o- verflow and contaminants invading, speed decided by sealing material	good sealing effect , can be tightened by bolt raising sealing pressure and compensating wear , large friction ,suit to low speed

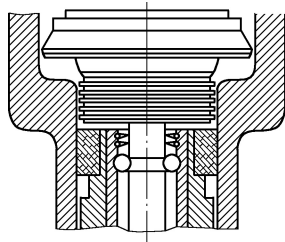
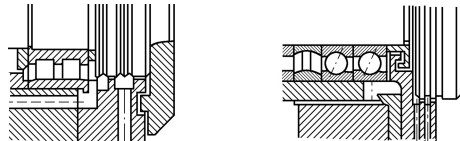
Table 3 – 60 Sealing purpose and structure (examples)

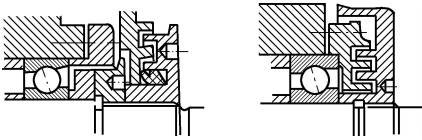
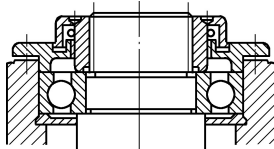
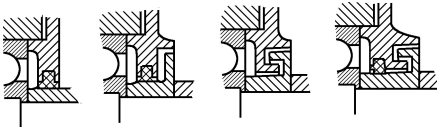
Sealing purpose										Typical condition	Structure example
prevent outside						prevent inside			high temperature		
solid				water		oil		grease			
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath				
✓									✓	motor	
	✓								✓	tow motor	

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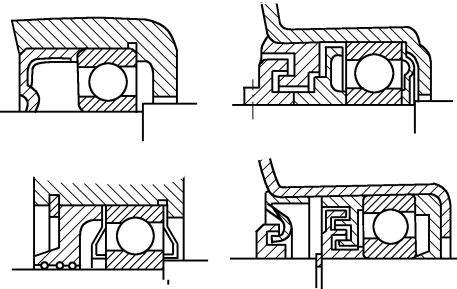
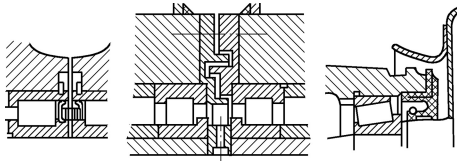
Sealing purpose										Typical condition	Structure example
prevent outside						prevent inside			high temperature		
solid				water		oil		grease			
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath				
			✓						✓	hammer grinding breaker, striking breaker	
			✓				✓			hammer breaker, hammer grinding breaker	
			✓						✓	converter	

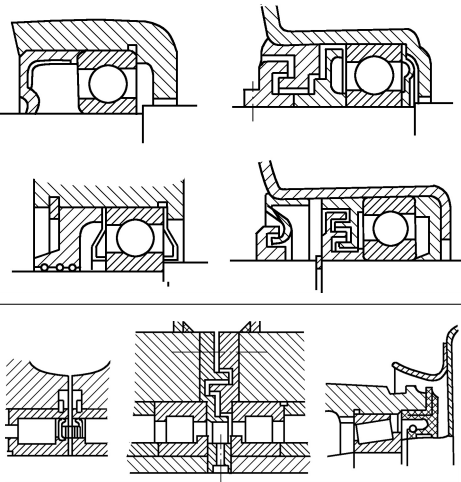
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Sealing purpose										high temperature	Typical condition	Structure example
prevent outside						prevent inside						
solid			water			oil		grease				
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath		circulate			
	✓								✓	spindle		
✓								✓		lathe spindle		

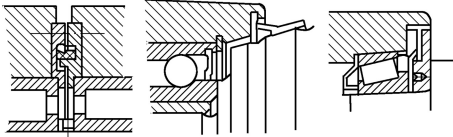
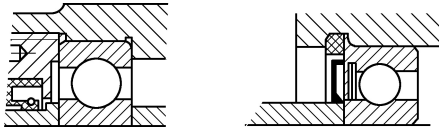
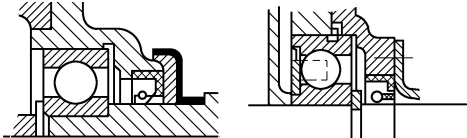
Sealing purpose											Typical condition	Structure example
prevent outside							prevent inside		high temperature			
solid				water			oil					
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath	circulate				
		✓						✓			emery wheel	
		✓							✓		stand milling cutter shaft	
	✓								✓		circle plate saw grinding machine	

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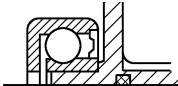
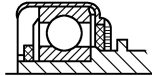
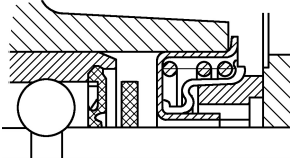
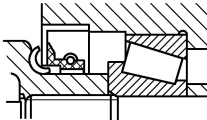
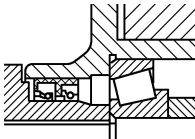
Sealing purpose										Typical condition	Structure example
prevent outside						prevent inside			high temperature		
solid			water			oil		grease			
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath				
			✓	✓					✓	conveyor belt roller	
✓					✓				✓	rope wheel	

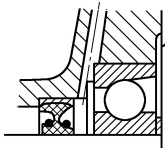
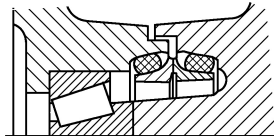
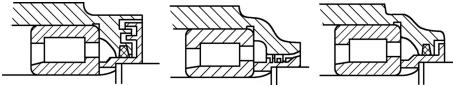


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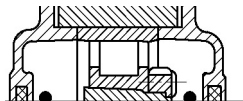
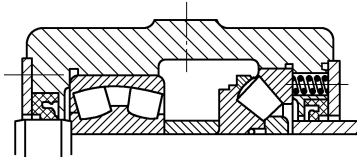
Sealing purpose										Typical condition	Structure example
prevent outside						prevent inside			high temperature		
solid				water		oil		grease			
small inf.	norminl inf.	large inf.	very large inf.	dripping water	spray water	high pre-ssue	oil bath				
		✓			✓				✓	automobile front wheel	
		✓			✓				✓	motorcycle hub	
		✓			✓		✓			automobile transmission	

(continued)

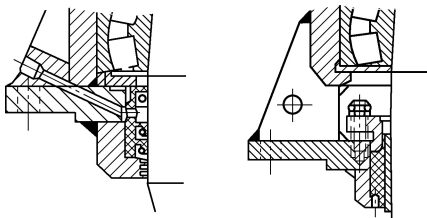
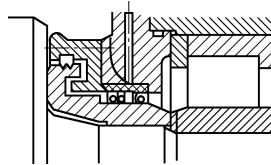
Sealing purpose										Typical condition	Structure example
prevent outside						prevent inside			high temperature		
solid			water			oil		grease			
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath				
	✓								✓	clutch separating bearing	 
✓						✓			✓	automobile water pump	
		✓			✓		✓			automobile gear bearing	 

Sealing purpose											Typical condition	Structure example
prevent outside							prevent inside		high temperature			
solid				water			oil			grease		
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath	circulate				
	sealing device inside										double-stroking engine	
			✓		✓		✓				excavator bearing	
		✓		✓					✓		railway vehicle car box bearing	

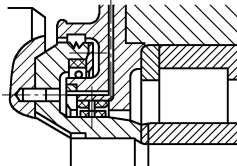
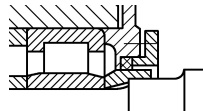
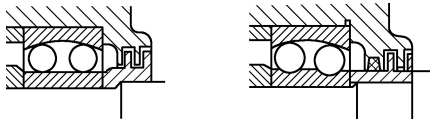
(continued)

Sealing purpose											Typical condition	Structure example
prevent outside						prevent inside			high temperature			
solid				water		oil		grease				
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath			circulate		
✓							✓				ship shaft support bearing	
✓					✓		✓				ship thrust support	

(continued)

Sealing purpose										Typical condition	Structure example	
prevent outside							prevent inside		high temperature			
solid				water			oil					grease
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath	circulate				
	✓	✓			✓	✓			✓ ✓	ship helm support W mode S mode		
sealing device prevent from milling lubrication oil											cool milling roller	

(continued)

Sealing purpose											Typical condition	Structure example
prevent outside						prevent inside			high temperature			
solid				water		oil		grease				
small inf.	normal inf.	large inf.	very large inf.	dripping water	spray water	high pressure	oil bath			circulate		
		✓			✓				✓	✓	Hot milling roller	
✓								✓		✓	rubber press machine	
	✓									✓	dry oven support	

3.6 Rolling Bearing Axial Fixation Design

3.6.1 Rolling Bearing Axial Locating and Fixing

The bearing axial fixation includes the axial locating and axial fixing. In order to prevent the bearing moving on shaft and in the housing, the bearing inner ring is fixed on the shaft; the bearing outer ring is fixed in the bearing housing (or in the sleeve). The support limiting position requirement and the bearing type are the main factors to decide whether the bearing inner ring and outer ring should be fixed on double directions or on single direction, or axial floating. Table 3-61 lists three kinds of axial fixation conditions of limiting position supports using different bearing types.

For the double direction limiting position support (locating support), both the bearing inner ring on shaft and bearing outer ring in housing should be fixed in double directions, as shown in Figure a to Figure e in Table 3-61. For the single direction limiting position support, there is merely need to fix the bearing inner ring and the outer ring in a single direction that is the opposing direction to the external axial load, such as shown in Figure f to Figure h in Table 3-61. If using the non-separable bearing as the floating support, there is merely need to fix the inner ring on both directions, the outer ring can displace freely in double axial directions relative to the housing bore. If using the separable radial bearing that the inner ring and the outer ring can be separated as the floating support, in this case, both the inner ring and the outer ring need to be fixed on the shaft and in the housing bore, the inner ring and the outer ring can make displacement relative to each other, such as shown in Figure i to Figure l in Table 3-61.

Generally, the axial locating of the bearing is that the inner ring is located by the shaft shoulder and the outer ring is by the housing bore resisting shoulder. In order to guarantee the locating function, the shaft shoulder and the housing bore resisting shoulder should be in closing contact with the end faces of the bearing inner ring and outer ring. For preventing

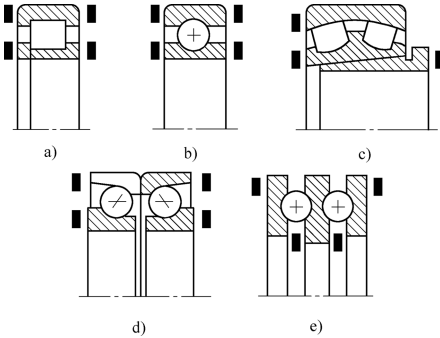
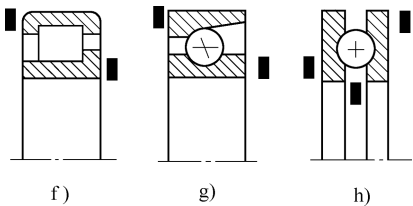
the interference, the relationship between the fillet radii (of the shaft shoulder and the housing resisting shoulder) and the relative bearing chamfer dimensions (of the inner ring and the outer ring) have to conform to the installing dimension stipulation that will be discussed later in this

chapter. The height of the lateral fixation of the shaft shoulder and the housing shoulder also must conform to the stipulation, so that the locating strength could be ensured and making the mounting and dismounting easily.

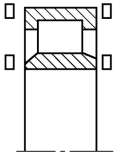
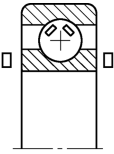
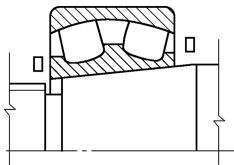
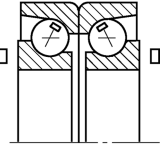
The fixation ways of the inner ring and the outer ring should accomodate the external axial load. The symbol “■” expresses the fixations subjected to considerable external axial loads, the symbol “□” expresses the fixations required merely to prevent axial displacement of the ring.

The examples of three kinds of fixations are listed in Table 3 - 1 , Table 3 - 2 and Table 3 - 3 in this chapter.

Table 3 - 61 Three limiting position support axial fixation sketch

Support type	Sketch	Illustration
double direction limiting Support (locating support)		subjected to double direction axial forces, need both direction axial fixation the fixation strength should accommodate the value of the force.
single directon limiting position support		subjected to only one direction axial force, merely need single direction fix- ation. Usually the clear- ance of two shaft ends can be adjustable.

(continued)

Support type	Sketch	Illustration
floating support		can be freely moving in axial direction, can eliminate the mounting error and compensate the axial heat expanding and cold shrinking amount, but can not subject to axial load, need no fixation strength, only need prevent bearing position displacement.
		
		
		

3.6.2 Several Commonly Used Axial Fixation Devices

There are many kinds of axial fixation device. Usually, the axial fixation device selection is decided according to the bearing type, axial load value, rotating speed (high or low), the bearing seat position on the shaft and the condition of mounting and dismounting. The axial fixation is required to be more reliable for heavier load and higher speed. For heavier axial load, usually using the locknut and lockwasher for the bearing inner ring axial fixation; using the end cap and the thread collar for the bearing outer ring axial fixation. For lighter load and lower rotating speed, usually using the shaft elastic locking ring, adapter sleeve and withdrawer sleeve etc. for the bearing inner ring axial fixation; using the bore elastic locking ring, snap ring etc. for the bearing outer ring axial fixation. Next we will introduce the commonly used axial fixation devices.

(1) Shaft abutment, end cap and locking ring

These devices are used for subjecting to radial load and alternating axial force, but the axial force is not so heavy and is not shocking force. The device making cost is low, and mounting and dismounting is very simple, such as the low power motor shaft supports.

(2) Locking nut and locking washer

The locking nut and the locking washer should be used at the same

time. The locking washer is used for preventing loose, and can be used for the conditions of heavier axial and radial load, such as the gear shaft supports, high speed machine tool spindle supports, wheel hub supports.

(3) Elastic snap ring

The elastic snap ring has two types: shaft using and bore using as shown in Figure 3 - 20. Their structures are very simple, and the dimensions are very small. They are suitable for the supports of ordinary rotating speed, usually heavy load.

(4) Adapter sleeve and withdrawal sleeve

1) Adapter sleeve

The adapter sleeve inner bore is a cylindrical surface fitting to the shaft; the outer surface is a tapered surface fitting to the tapered bearing bore. The adapter sleeve has been cut an axial slot along the axial direction, having elasticity. The tail part is a cylindrical surface with thread of screw for bearing axial fixation by locknut. This kind of fixation device has the advantages of simple structure and easily mounting and dismounting. The adapter sleeve is suitable specially for the cylindrical shaft that the rotating speed is not very high, and the axial load is not very heavy.

2) Withdrawal sleeve

The withdrawal sleeve inner bore is a cylindrical surface fitting to the shaft; the outer surface is a tapered surface fitting to the tapered bearing bore. The withdrawal sleeve has been cut an axial slot along the axial direction. The withdrawal sleeve can be pushed into the bearing bore by using shaft locknut. The tail part is a cylindrical surface with thread of screw for dismounting. The withdrawal sleeve can be self-locked, this kind of fixation device has the advantages of simple structure and easily mounting and dismounting. The withdrawal sleeve is self locked, suitable specially for elevating bearing axial fixation of rope wheel, but it is also suitable for the self-aligning bearing mounting in cylindrical shaft that the rotating speed is not very high, and the axial load is not very heavy.

(5) Elastic snap ring

The bearings with circular groove and elastic snap ring can be easily

fixed in the housing. Such bearings are generally used in automobile transmissions where, due to the limited space, the outer ring must be fixed in a narrow housing wall. Because of the low manufacturing cost, this type of fixation is particularly suitable for series production.

(6) Locating ring and eccentric locking collar

Bearing housing with series production can be designed as not only the housing of axial locating bearing, but also the housing of axial floating bearing in order to easily store. The housing fitting dimensions can be widened, so if mounting the axial floating bearing, it can be suitable for mounting the maximum with bearing of diameter series. If mounting one or two locating rings, that will become a locating bearing. These locating rings can supply to every width series bearing for using.

The insert bearing with eccentric locking collar can be fixed by the eccentric locking collar and locating ring.

(7) Flange

In special condition we can use bearing outer ring with flange for fixing.

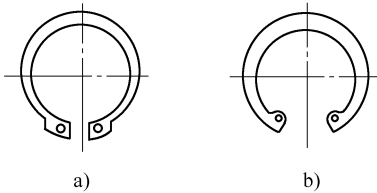


Figure 3-20 Elastic snap ring

- a) shaft snap ring
- b) housing snap ring

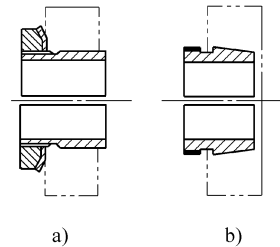


Figure 3-21 Adapter sleeve and withdrawal sleeve

- a) adapter sleeve
- b) withdrawal sleeve

3.6.3 Common Bearing Inner Ring and Outer Ring Fixation Device

Table 3-62 lists the common bearing inner ring fixation device. Table 3-63 lists the common bearing outer ring fixation device.

Table 3 – 62 Common bearing inner ring fixation device

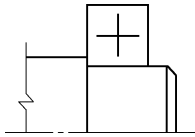
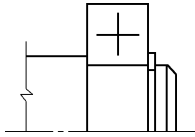
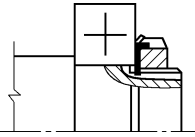
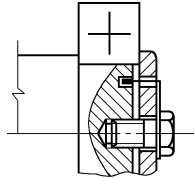
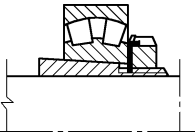
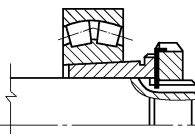
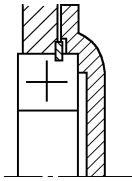
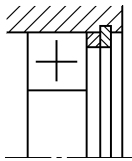
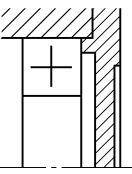
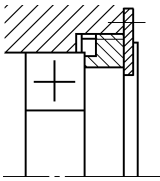
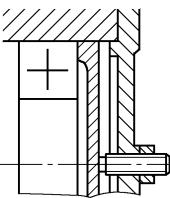
Order	Name	Sketch	Illustration
1	shaft abutment fixation		bearing inner ring relies on shaft abutment and interference fit to get the axial fixation. Suitable for two end locating support structure. Simple structure and smaller outline dimension
2	shaft abutment elastic snap ring fixation		bearing inner ring relies on the shaft abutment and elastic snap ring to get the axial fixation. Can subject to double direction axial load of little value, smaller axial structure dimension
3	locknut fixation		bearing inner ring relies on shaft abutment and locknut to get the axial fixation and having the snap washer preventing loose, safety and reliability, suitable for high speed and heavy load
4	end face thrust washer fixation		Bearing inner ring relies on shaft abutment and end face washer to get axial fixation. The end face washer fixed on the end face by screw. The fixing screw should have preventing loosen device. Suitable for the applications that can not make thread screw on the end face or the space is limited
5	adapter sleeve fixation		Relying on pressing the radial dimension of the adapter inner bore to clamp on to the shaft for getting the axial fixation of the bearing inner ring
6	withdrawal sleeve fixation		The clamping way of the withdrawal sleeve is the same as the the adapter sleeve. Because of the special screw nut of the withdrawal sleeve, mounting and dismantling of the bearing is very easy.

Table 3-63 Common bearing outer ring fixation device

Order	Name	Sketch	Illustration
1	snap ring fixation		Radial ball bearing outer ring with the snap ring groove can use this kind of fixation. The structure is very simple, the axial dimension is smaller. But this kind of fixation can not subject to heavy axial load.
2	elastic snap ring fixation		Simple structure, easily mounting and dismounting, small axial dimension. Can add adjustment ring between the bearing end face and the snap ring, also can adjust the bearing axial position compensating the manufacturing and the mounting errors. Can use in the conditions of low speed and light axial load
3	end cap fixation		Using for radial and angular contact ball bearing fixation on shaft end. The cap can be made as various forms. When the cap bore is a through bore, it also can carry various sealingdevice. Suitable for the conditions of high speed and heavy axial load.
4	screw ring fixation		Suitable for the conditions of high speed and heavy axial load. The screw ring also can adjust the clearance of face to face arrangement angular contact bearing. The screw ring should have the anti-loosen measure.
5	adjusting screw and adjust cap fixation		Similar to end cap fixation. This device makes the bearing clearance adjustment convenient. The adjustment screw should have the anti-loosen measure.

3.7 Rolling Bearing Installation Dimensions

The rolling bearing installation dimensions, that is the dimensions related to the shaft and the housing of the mounted bearing, are very important for designing the components matching to the bearing. The China National Standard GB274—2000 and GB5868—2003 stipulated the mounting chamfers and the rolling bearing installation dimensions.

3.7.1 Dimension Limit of rolling Bearing Mounting Chamfer

For metric system bearing rings, GB274—2000 stipulated the mounting chamfer dimension terms, symbols and the dimension limit and the fillet radii of the shaft and the housing. Figure 3-22 is the sketch. There is no stipulation about the exact chamfer shape, however, the chamfer contour in the axial plane can not exceed the imaginary circle arc of radius r_{smmin} formed by the tangents of the ring end face and the inner bore or the outside cylindrical surface. The meaning of the symbols are as follows:

d —nominal bearing bore diameter;

D —nominal bearing outer diameter;

r_s —radial bearing, tapered roller bearing and thrust single chamfer dimension of bearing;

r_{1s} —single chamfer dimension of cylindrical roller bearing loose rib and thrust collar and one snap groove side outer ring; single chamfer dimension of cylindrical roller bearing narrow end faces of inner ring and outer ring and angular contact ball bearing outer ring narrow end face; single chamfer dimension of thrust bearing guide ring;

r_{as} —single chamfer dimension of shaft and housing;

r_{smin} , $r_{1\text{smin}}$ —permissible minimum single chamfer dimension of r_s or r_{1s} ;

r_{smax} , $r_{1\text{smax}}$ —permissible maximum single chamfer dimension of r_s or r_{1s} ;

r_{amax} —permissible maximum single chamfer dimension of shaft or housing bore.

Note: There is no stipulation about the exact chamfer shape, however, the chamfer contour in the axial plane can not exceed the imaginary circle arc of radius r_{smin} formed by the tangents of the ring end face and the inner bore or the outside cylindrical surface. See Figure 3-22.

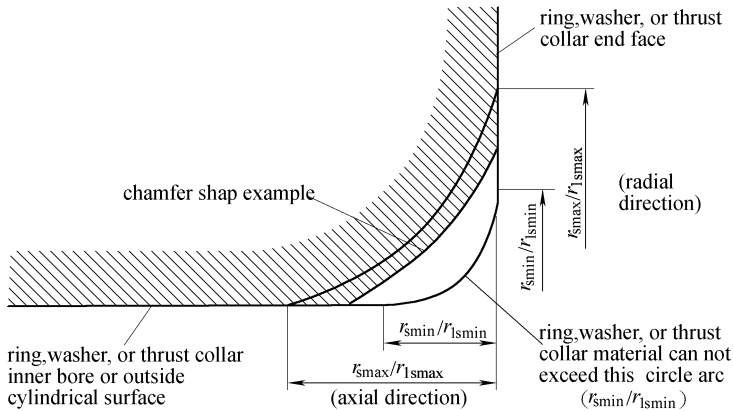


Figure 3-22 Mounting chamfer sketch

Bearing chamfer dimension is shown in Table 3-64 to Table 3-68.

Table 3-64 Radial bearing chamfer dimension maximum value

(unit: mm)

r_{smin} ①	Radial bearing chamfer dimension maximum value			
	d		r_{smax} ②	
	over	to	radial direction	axial direction
0.05	—	—	0.1	0.2
0.08	—	—	0.16	0.3
0.1	—	—	0.2	0.4
0.15	—	—	0.3	0.5
0.2	—	—	0.5	0.6
0.3	—	40	0.6	1
	40	—	0.8	1
0.6	—	40	1	2
	40	—	1.3	2
1	—	50	1.5	3
	50	—	1.9	3
1.1	—	120	2	3.5
	120	—	2.5	4
1.5	—	120	2.3	4
	120	—	3	5

(continued)

$r_{\text{smin}}^{①}$	Radial bearing chamfer dimension maximum value			
	d		$r_{\text{smax}}^{②}$	
	over	to	radial direction	axial direction
2	—	80	3	4.5
	80	220	3.5	5
	220	—	3.8	6
2.1	—	280	4	6.5
	280	—	4.5	7
2.5 ^③	—	100	3.8	6
	100	280	4.5	6
	280	—	5	7
3	—	280	5	8
	280	—	5.5	8
4	—	—	6.5	9
5	—	—	8	10
6	—	—	10	12
7.5	—	—	12.5	17
9.5	—	—	15	19
12	—	—	18	24
15	—	—	21	30
19	—	—	25	38

- ① Maximum single fillet radius dimensions of shaft and housing are in section 7.2 of this chapter.
- ② For the bearing width $\leq 2\text{mm}$, the radial value is also suitable for axial direction.
- ③ No stipulating value of this chamfer dimension in GB/T 273.3.

maximum single chamfer dimension of cylindrical roller bearing loose rib and thrust collar and one snap groove side outer ring

$r_{\text{lmin}}^{①}$	d or D		r_{lmax}	
	over	to	radial direction	axial direction
0.2	—	—	0.5	0.5
0.3	—	40	0.6	0.8
	40	—	0.8	0.8
0.5	—	40	1	1.5
	40	—	1.3	1.5
0.6	—	40	1	1.5
	40	—	1.3	1.5
1	—	50	1.5	2.2
	30	—	1.8	2.2
1.1	—	120	2	2.7
	120	—	2.5	2.7

(continued)

maximum single chamfer dimension of cylindrical roller bearing narrow end faces of inner ring and outer ring and angular contact ball bearing outer ring narrow end face

$r_{1\min}$ ①	d or D		$r_{1\max}$	
	over	to	radial direction	axial direction
1.5	—	120	2.3	3.5
	120	—	3	3.5
2	—	80	3	4
	80	220	3.5	4
	220	—	3.8	4
2.1	—	280	4	4.5
	280	—	4.5	4.5
2.5 ②	—	100	3.8	5
	100	280	4.5	5
	280	—	5	5
3	—	280	5	3.5
	280	—	5.5	5.5
4	—	—	6.5	6.5
5	—	—	8	8
6	—	—	10	10

① Maximum single fillet radius dimensions of shaft and housing are in section 7.2 of this chapter.

② No stipulating value of this chamfer dimension in GB/T 283 and GB/T 305.

maximum narrow end face chamfer dimension of cylindrical roller bearing inner ring and outer ring and angular contact ball bearing outer ring

$r_{1\min}$ ①	d or D		$r_{1\max}$	
	over	to	radial direction	axial direction
0.1	—	—	0.2	0.1
0.15	—	—	0.3	0.6
0.2 ②	—	—	0.5	0.8
0.3	—	40	0.6	1
	40	—	0.8	1
0.6	—	40	1	2
	40	—	1.3	2
1	—	50	1.5	3
	50	—	1.9	3
1.1	—	120	2	3.5
	120	—	2.5	4
1.5	—	120	2.3	4
	120	—	3	5

(continued)

r_{1min} ①	d or D		r_{1max}	
	over	to	radial direction	axial direction
2	—	80	3	4.5
	80	220	3.5	5
	220	—	3.8	6

- ① Maximum single fillet radius dimensions of shaft and housing are in section 7.2 of this chapter.
- ② No stipulating value of this chamfer dimension in GB/T 283 and GB/T 292.

Table 3 – 65 Dimension limit of tapered roller bearing chamfer

(unit: mm)

inner ring (d) or outer ring (D) large end face chamfer				
r_{smin} ①	d or D		r_{smax}	
	>	≤	radial	axial
0.3	—	40	0.7	1.4
	40	—	0.9	1.6
0.6	—	40	1.1	1.7
	40	—	1.3	2
1.0	—	50	1.6	2.5
	50	—	1.9	3
1.5	—	120	2.3	3
	120	250	2.8	3.5
	250	—	3.5	4
2	—	120	2.8	4
	120	250	3.5	4.5
	250	—	4	5
2.5	—	120	3.5	5
	120	250	4	5.5
	250	—	4.5	6
3	—	120	4	5.5
	120	250	4.5	6.5
	250	400	5	7
	400	—	5.5	7.5
4	—	120	5	7
	120	250	5.5	7.5
	250	400	6	8
	400	—	6.5	8.5
5	—	180	6.5	8
	180	—	7.5	9

(continued)

inner ring (<i>d</i>) or outer ring (<i>D</i>) large end face chamfer				
r_{min} ①	<i>d</i> or <i>D</i>		r_{smax}	
	>	≤	radial	axial
6	—	180	7.5	10
	180	—	9	11

① Maximum single fillet radius dimensions of shaft and housing are in section 7.2 of this chapter.

Table 3 – 66 Thrust bearing chamfer maximum value

(unit: mm)

r_{smin} ① or r_{lsmin} ①	r_{smax} or r_{lsmax}	r_{smin} ① or r_{lsmin} ①	r_{smax} or r_{lsmax}
	radial or axial		radial or axial
0.3	0.8	4	6.5
0.6	1.5	5	8
1	2.2	6	10
1.1	2.7	7.5	12.5
1.5	3.5	9.5	15
2	4	12	18
2.1	4.5	15	21
3	5.5	19	25

Note: the values in Table are suitable for:

- 1) The housing washer bottom plane and the outside cylindrical surface chamfers;
- 2) The single direction shaft washer bottom plane and bore surface chamfers;
- 3) Double direction bearing middle ring end face and bore surface chamfer.

① Maximum single fillet radius dimensions of shaft and housing are in section 3.7.2 of this chapter.

Table 3 – 67 Radial bearing nominal chamfer dimension and single direction minimum chamfer dimension contrast table (unit: mm)

<i>r</i> (nominal chamfer dimension)	0.1	0.15	0.2	0.3	0.4	0.5	1	1.5	2	2.5	3
r_{smin}	0.05	0.08	0.1	0.15	0.2	0.3	0.6	1	1.1	1.5	2
<i>r</i> (nominal chamfer dimension)	3.5	4	5	6	8	10	12	15	18	22	
r_{smin}	2.1	3	4	5	6	7.5	9.5	12	15	19	

Table 3 – 68 Tapered roller bearing nominal chamfer dimension and end face minimum chamfer dimension contrast table

r (nominal)	inner ring large end chamfer	outer ring large end chamfer
	r_{smin}	r_{smin}
0.5	0.3	0.3
1	0.6	0.6
1.5	1	1
2	1.5	1.5
2.5	2	1.5
3	2.5	2
3.5	3	2.5
4	4	3
5	5	4
6	6	5

3. 7. 2 Single Direction Fillet Radius of Shaft and Housing

In order to ensure the bearing being in closing contact with the shaft shoulder and the housing shoulder, to ensure the bearing end face contacting with the shaft shoulder and the housing shoulder, to prevent the interim fillet radius from colliding with the bearing chamfer (see Figure 3 – 23, 3 – 24), the maximum single direction fillet radius of the shaft and housing bore must conform to the stipulation in Table 3 – 69. The permissible maximum single fillet radius r_{asmax} of the shaft and the housing should not be larger than the permissible minimum single chamfer dimension r_{smin} and r_{lsmin} of the corresponding ring or washer.

The h shown in Figures 3 – 23 and 3 – 24 is the shoulder height of the shaft or the housing.

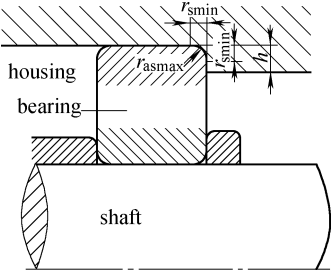


Figure 3 – 23 Housing bore fillet radius

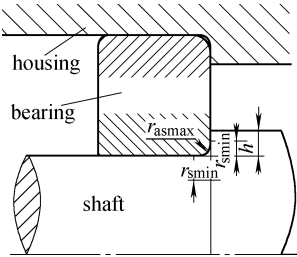


Figure 3 – 24 Shaft shoulder fillet radius

Table 3 – 69 Single maximum fillet radius r_{asmax} of shaft and housing

(unit: mm)

Bearing minimum single chamfer dimension r_{smin}	r_{asmax} max	Bearing minimum single chamfer dimension r_{smin}	r_{asmax} max
0.05	0.05	2	2
0.08	0.08	2.1	2.1
0.1	0.1	3	2.5
0.15	0.15	4	3
0.2	0.2	5	4
0.3	0.3	6	3
0.6	0.6	7.5	6
1	1	9.5	8
1.1	1	12	10
1.5	1.5	15	12

3.7.3 Shoulder Height of Deep Groove Ball Bearings, Angular Contact Ball Bearings, Self-Aligning Ball Bearings and Self-Aligning Roller Bearings

When determining the shoulder height, firstly we consider its closing contact with the bearing end face, then to consider its easiness using the mounting and dismounting tool. Table 3 – 70 stipulates the shoulder minimum height.

Table 3 – 70 Shoulder minimum height (unit: mm)

Bearing minimum single chamfer dimension r_{smin}	h_{min}		Bearing minimum single chamfer dimension r_{smin}	h_{min}	
	usual condition	special condition ^①		usual condition	special condition
0.05	0.2	—	0.15	0.6	—
0.08	0.3	—	0.2	0.8	—
0.1	0.4	—	0.3	1.2	1
0.6	2.5	2	4	9	8
1	3	2.5	5	11	10
1.1	3.5	3.3	6	14	12
1.5	4.5	4	7.5	18	—

(continued)

Bearing minimum single chamfer dimension $r_{\min}$	$h_{\min}$		Bearing minimum single chamfer dimension $r_{\min}$	$h_{\min}$	
	usual condition	special condition ^①		usual condition	special condition
2	5	4.5	9.5	22	—
2.1	6	5.5	12	27	—
3	7	6.5	15	32	—

① The special condition designates the very small thrust load condition , or the condition of having a requirement that the shoulder must be very small on designing.

3.7.4 Installation Dimensions of Cylindrical Roller Bearings

The cylindrical roller bearing is a kind of separable bearing, its inner ring and outer ring can be installed on the shaft and the housing separately, therefore, during installation, we merely need to push the entire bearing into the housing.

However, in order to ensure that the bearing matching component and sealing device would not obstruct the support installation, and for mounting and dismounting conveniently, the cylindrical roller bearing installation dimension must conform to the stipulation in Table 3 - 71 and Table 3 - 72.

Table 3 - 71 lists the 10 series, 02 series cylindrical roller bearing installation dimensions, and Table 3 - 72 lists the the 03 series, 04 series cylindrical roller bearing installation dimensions. Figure 3 - 25 to Figure 3 - 28 are the cylindrical roller bearing installation dimensions.

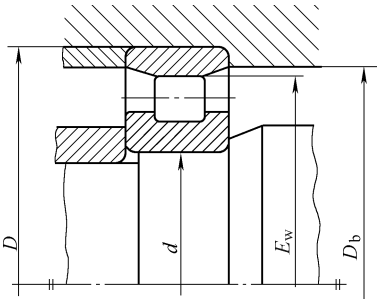


Figure 3 - 25 N type cylindrical roller bearing installation dimension

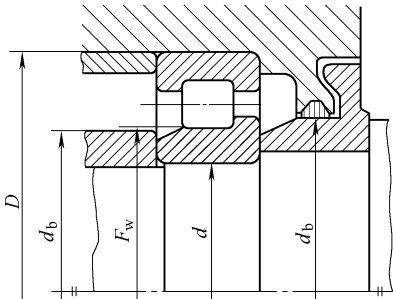


Figure 3 - 26 NJ type cylindrical roller bearing installation dimension

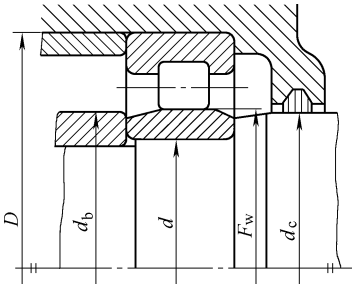


Figure 3-27 NU type cylindrical roller bearing installation dimension

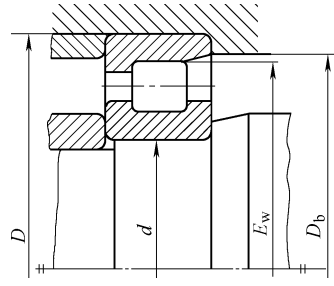


Figure 3-28 NF type cylindrical roller bearing installation dimension

In Figure 3-25 to Figure 3-28:

B_a , B_b —spacer inner ring width;

D —bearing nominal outer ring diameter;

D_a —housing bore shoulder diameter;

D_b —outer ring, snap ring inner ring diameter or housing bore shoulder diameter;

d —bearing nominal bore diameter;

d_a —shaft shoulder diameter;

d_b , d_c , d_d —inner ring, snap ring, spacer ring outer diameter or shaft, inner bore diameter;

d_1 —adapter inner diameter;

E_w —roller (needle) set outer ring diameter;

F_w —roller (needle) set inner ring diameter;

h —shaft and housing shoulder height;

N_b , N_t —spacer ring width;

$r_{\min}$ —bearing minimum single chamfer dimension;

$r_{\max}$ —shaft and housing maximum chamfer dimension;

S_a —recess width that is opposite to the cage using for installing the tapered roller bearing outer ring wide face;

S_b —recess width that is opposite to the cage using for installing the tapered roller bearing outer ring narrow face;

Table 3 – 71 10 series, 02 series cylindrical roller bearing installation dimensions

(unit: mm)

d	10 series				02 series						
	D	NU1000E			D	NU200E NJ200E		NU200E	NJ200E	N200E NF200E	
		F_w	d_b	d_c		d_c	d_d	E_w	D_b		
			max	min						min	min
20	—	—	—	—	47	26.5	26	29	32	41.5	42
25	47	30.5	30	32	52	31.5	31	34	37	46.5	47
30	55	36.5	35	38	62	37.5	37	40	44	55.5	56
35	62	42	41	44	72	44	43	46	50	64	64
40	68	47	46	49	80	49.5	49	52	56	71.5	72
45	75	52.5	52	54	85	54.5	54	57	61	76.5	77
50	80	57.5	57	59	90	59.5	58	62	67	81.5	83
55	90	64.5	63	66	100	66	65	68	73	90	91
60	95	69.5	68	71	110	72	71	75	80	100	100
65	100	74.5	73	76	120	78.5	77	81	87	108.5	109
70	110	80	78	82	125	83.5	82	86	92	113.5	114
75	115	85	83	87	130	88.5	87	90	96	118.5	120
80	125	91.5	90	94	140	95.3	94	97	104	127.3	128
85	130	96.5	95	99	150	100.5	99	104	110	136.5	137
90	140	103	101	106	160	107	105	110	116	145	146
95	145	108	106	111	170	112.5	111	116	123	154.5	155
100	150	113	111	116	180	119	117	122	130	163	164
105	160	119.5	118	122	190	125	124	130	137	173	173
110	170	125	124	128	200	132.5	130	137	144	180.5	182
120	180	135	134	138	215	143.5	141	144	156	195.5	196
130	200	148	146	151	230	153.5	151	158	168	209.5	208
140	210	158	156	161	250	169	166	171	182	225	225
150	225	169.5	167	173	270	182	179	184	196	242	242
160	240	180	178	184	290	195	192	197	210	259	261
170	260	193	190	197	310	207	204	211	223	279	280
180	280	205	203	209	320	217	214	221	233	289	290
190	290	215	213	219	340	230	227	234	247	306	307
200	310	229	226	233	360	243	240	247	261	323	323
220	340	250	248	254	400	268	266	273	289	—	—
240	360	270	268	275	440	293	290	298	316	—	—
260	400	296	292	300	480	317	315	323	343	—	—

Table 3 – 72 03 series, 04 series cylindrical roller bearing installation dimensions

(unit: mm)

d	03 series							04 series				
	D	NU300E NJ300E		NU300E	NJ300E	N300E NF300E		D	NU400 NJ400		NU400	NJ400
		F_w	d_b max	d_c min	d_d min	E_w	D_b min		F_w	d_b max	d_c min	d_d min
20	52	27.5	27	30	33	45.4	47	—	—	—	—	—
25	62	34	33	37	40	54	55	—	—	—	—	—
30	72	40.5	40	44	48	62.6	64	90	45	44	47	52
35	80	46.2	45	48	53	70.2	71	100	53	52	55	61
40	90	52	51	55	60	80	80	110	58	57	60	67
45	100	58.5	57	60	66	88.5	89	120	64.5	63	66	74
50	110	65	63	67	73	97	98	130	70.8	69	73	81
55	120	70.5	69	72	80	106.5	107	140	77.2	76	79	87
60	130	77	75	79	86	115	116	150	83	82	85	94
65	140	82.5	81	85	93	124.5	125	160	89.5	88	91	100
70	150	89	87	92	100	133	134	180	100	99	102	112
75	160	95	93	97	106	143	143	190	104.5	103	107	118
80	170	101	99	105	114	151	151	200	110	109	112	124
85	180	108	106	110	119	160	160	210	115.5	111	115	128
90	190	113.5	111	117	127	169.5	170	225	123.5	122	125	139
95	200	121.5	119	124	134	177.5	178	240	133.5	132	136	149
100	215	127.5	125	132	143	191.5	192	250	139	137	141	156
105	225	133	132	137	149	201	201	260	144.5	143	147	162
110	240	143	140	145	158	211	211	280	155	153	157	173
120	260	154	151	156	171	230	230	310	170	168	172	190
130	280	167	164	169	184	247	247	340	185	183	187	208
140	300	180	176	182	198	260	260	360	196	195	200	222
150	320	193	190	195	213	283	283	380	209	208	216	237
160	370	204	200	211	228	300	300	—	—	—	—	—
170	360	218	216	223	241	318	318	—	—	—	—	—
180	380	231	227	235	255	—	—	—	—	—	—	—
190	400	245	240	248	268	—	—	—	—	—	—	—
200	420	258	254	263	283	—	—	—	—	—	—	—

3. 7. 5 Installation Dimensions of Needle Roller Bearings

The solid needle roller bearings are separable type, which installation dimensions are shown in Figure 3 – 29 and Table 3 – 73 for convenient mounting and dismounting.

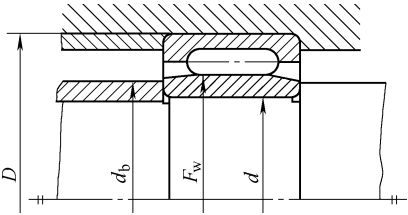


Figure 3 – 29 Solid needle roller bearing installation dimension

Table 3 – 73 Solid outer ring needle roller bearing installation dimension

(unit: mm)

d	N A 48 series			N A 48 series		
	D	F_w	d_b max	D	F_w	d_b max
15	—	—	—	28	20	19
17	—	—	—	30	22	21
20	—	—	—	37	25	21
22	—	—	—	39	28	27
25	—	—	—	42	30	29
28	—	—	—	45	32	31
30	—	—	—	47	35	34
32	—	—	—	52	40	39
35	—	—	—	55	42	41
40	—	—	—	62	48	47
45	—	—	—	68	52	51
50	—	—	—	72	58	57
55	—	—	—	80	63	61
60	—	—	—	85	68	66
65	—	—	—	90	72	70
70	—	—	—	100	80	78
75	—	—	—	105	85	83
80	—	—	—	110	90	88
85	—	—	—	120	100	98

(continued)

d	NA48 series			NA48 series		
	D	F_w	d_b max	D	F_w	d_b max
90	—	—	—	125	105	103
95	—	—	—	130	110	108
100	—	—	—	140	115	113
110	140	120	118	150	125	123
120	150	130	128	165	135	133
130	165	145	143	180	150	148
140	175	155	153	190	160	158
150	190	165	163	—	—	—
160	200	175	173	—	—	—
170	215	185	183	—	—	—
180	225	195	193	—	—	—
190	240	210	203	—	—	—
200	250	220	218	—	—	—

3.7.6 Installation Dimensions of Tapered Roller Bearings

The tapered roller bearing cage protrudes beyond the outer ring wide end face, as shown in Figure 3-30. In order to prevent the cage from touching the housing and to prevent other matching components from touching the cage during installation, the tapered roller bearing installation dimensions must conform to the values listed in Table 3-74.

Table 3-74 lists the 02 series tapered roller bearing installation dimensions.

Table 3-75 lists the 03 series tapered roller bearing installation dimensions.

Table 3-76 lists the 13 series tapered roller bearing installation dimensions.

Table 3-77 lists the 20 series tapered roller bearing installation

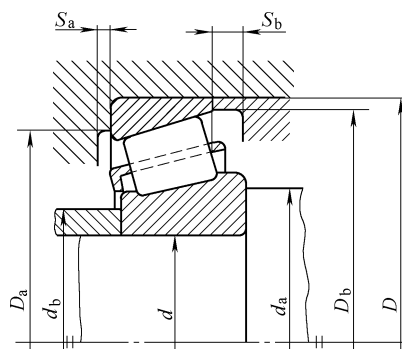


Figure 3-30 Tapered roller bearing installation dimension

dimensions.

Table 3 – 78 lists the 22 series tapered roller bearing installation dimensions.

Table 3 – 79 lists the 23 series tapered roller bearing installation dimensions.

Table 3 – 74 02 series tapered roller bearing installation dimension

(unit: mm)

<i>d</i>	302 series							
	<i>D</i>	<i>d_a</i> min	<i>d_b</i> max	<i>D_a</i>		<i>D_b</i> min	<i>S_a</i> min	<i>S_b</i> min
				min	max			
17	40	23	23	34	34	37	2	2.5
20	47	26	27	40	41	43	2	3.5
25	52	31	31	44	46	48	2	3.5
30	62	36	37	53	56	57	2	3.5
32	65	38	39	56	59	60	3	3.5
35	72	42	44	62	65	67	3	3.5
40	80	47	49	69	73	74	3	4.0
45	85	52	54	74	78	80	3	5.0
50	90	57	58	79	83	85	3	5.0
55	100	64	64	88	91	94	4	5.0
60	110	69	70	96	101	103	4	5.0
65	120	74	77	106	111	113	4	5.0
70	125	79	81	110	116	118	4	5.5
75	130	84	86	115	121	121	4	5.5
80	140	90	91	124	130	133	4	6.0
85	150	95	97	132	140	141	5	6.5
90	160	100	103	140	150	151	5	6.5
95	170	107	109	149	158	160	5	7.5
100	180	112	115	157	168	169	5	8.0
105	190	117	122	165	178	178	6	9.0
110	200	122	129	174	188	188	6	9.0
120	215	132	140	187	203	202	6	9.5
130	230	144	152	203	216	218	7	10.0

(continued)

d	302 series							
	D	d_a min	d_b max	D_a		D_b min	S_a min	S_b min
				min	max			
140	250	154	163	219	236	235	9	11.0
150	270	164	175	234	256	251	9	11.0
160	290	174	189	252	276	270	9	12.0
170	310	188	201	269	292	290	9	14.0
180	320	198	209	278	302	299	9	14.0
190	340	208	223	298	322	320	9	14.0
200	360	218	235	315	342	338	9	16.0

Table 3 – 75 03 series tapered roller bearing installation dimension

(unit: mm)

d	303 series							
	D	d_a min	d_b max	D_a		D_b min	S_a min	S_b min
				min	max			
15	42	21	22	36	36	38	2	3.5
17	47	23	25	40	41	42	3	3.5
20	52	27	28	44	45	47	3	3.5
25	62	32	35	54	55	57	3	3.5
30	72	37	41	63	65	66	3	5.0
35	80	44	45	70	71	74	3	5.0
40	90	49	52	77	81	82	3	5.5
45	100	54	59	86	91	92	3	5.5
50	110	60	65	95	100	102	4	6.5
55	120	65	71	104	110	112	4	6.5
60	130	72	77	112	118	121	5	7.5
65	140	77	83	122	128	131	5	8.0
70	150	82	89	130	138	140	5	8.0
75	160	87	95	139	148	149	5	9.0
80	170	92	102	148	158	159	5	9.5
85	180	99	107	156	168	168	6	10.5

(continued)

<i>d</i>	303 series							
	<i>D</i>	<i>d_a</i> min	<i>d_b</i> max	<i>D_a</i>		<i>D_b</i> min	<i>S_a</i> min	<i>S_b</i> min
				min	max			
90	190	104	113	165	176	177	6	10.5
95	200	109	118	172	186	185	6	11.5
100	215	114	127	184	201	198	6	12.5
105	225	119	133	193	211	207	7	12.5
110	240	124	142	206	226	221	8	12.5
120	260	134	153	221	246	238	8	13.5
130	280	145	164	239	262	257	8	15.0
140	300	155	176	255	282	275	9	15.0
150	320	165	189	273	302	294	9	17.0
160	340	175	201	290	322	312	9	17.0
170	360	185	214	307	342	331	10	18.0
180	380	198	228	327	362	351	10	19.0

Table 3 – 76 13 series tapered roller bearing installation dimension

(unit: mm)

<i>d</i>	313 series							
	<i>D</i>	<i>d_a</i> min	<i>d_b</i> max	<i>D_a</i>		<i>D_b</i> min	<i>S_a</i> min	<i>S_b</i> min
				min	max			
25	62	32	33	47	55	59	3	5.5
30	72	37	38	55	65	68	3	7.0
35	80	44	48	62	71	76	4	8.0
40	90	49	49	71	81	86	4	8.5
45	100	54	55	79	91	95	4	9.5
50	110	60	60	87	100	105	4	10.5
55	120	65	64	94	110	114	4	10.5
60	130	72	70	103	118	124	5	11.5
65	140	77	76	111	128	133	5	13.0
70	150	82	81	118	138	142	5	13.0
75	160	87	87	127	148	153	6	14.0
80	170	92	92	134	158	160	6	15.5

(continued)

d	313 series							
	D	d_a min	d_b max	D_a		D_b min	S_a min	S_b min
				min	max			
85	180	99	98	143	166	170	6	16.5
90	190	104	103	151	176	180	6	16.5
95	200	109	108	157	186	188	6	17.5
100	215	114	117	168	201	203	7	21.5
105	225	119	122	176	211	212	7	22.0
110	240	124	132	188	226	225	7	25.0
120	260	134	141	203	246	245	9	26.0
130	280	147	151	218	262	263	9	28.0
140	300	157	163	235	282	282	9	30.0
150	350	167	174	251	302	302	9	32.0

Table 3 – 77 20 series tapered roller bearing installation dimension

(unit: mm)

d	320 series							
	D	d_a min	d_b max	D_a		D_b min	S_a min	S_b min
				min	max			
28	52	34	33	45	46	49	3	4.0
30	55	36	35	48	49	52	3	4.0
32	58	38	38	50	52	55	3	4.0
35	62	41	40	54	56	59	4	4.0
40	68	46	46	60	62	65	4	4.5
45	75	51	51	67	69	72	4	4.5
50	80	56	56	72	74	77	4	4.5
55	90	62	63	81	83	86	4	5.5
60	95	67	67	85	88	91	4	5.5
65	100	72	72	90	93	97	4	5.5
70	110	77	78	98	103	105	5	6.0
75	115	82	83	103	108	110	5	6.0
80	125	87	89	112	117	120	6	7.0
85	130	92	94	117	122	125	6	7.0

(continued)

<i>d</i>	320 series							
	<i>D</i>	<i>d</i> _a min	<i>d</i> _b max	<i>D</i> _a		<i>D</i> _b min	<i>S</i> _a min	<i>S</i> _b min
				min	max			
90	140	99	100	125	131	134	6	8.0
95	145	104	105	130	136	140	6	8.0
100	150	109	109	134	141	144	6	8.0
105	160	115	116	143	150	154	6	9.0
110	170	120	122	152	160	163	7	9.0
120	180	130	131	161	170	173	7	9.0
130	200	140	144	178	190	192	8	11.0
140	210	150	153	187	200	202	8	11.0
150	225	162	164	200	213	216	8	12.0
160	240	172	175	213	228	231	8	13.0
170	260	182	187	230	248	249	10	14.0
180	280	192	199	247	268	267	10	16.0
190	290	202	209	257	278	279	10	16.0
200	310	212	221	273	298	297	11	17.0
220	340	234	243	300	326	326	12	19.0
240	360	254	261	318	346	346	12	19.0
260	400	278	287	352	382	383	14	22.0
280	420	298	305	370	402	402	14	22.0
300	460	318	329	404	442	439	15	26.0
320	480	338	350	424	462	461	15	26.0

Table 3 – 78 22 series tapered roller bearing installation dimension

(unit: mm)

<i>d</i>	322 series							
	<i>D</i>	<i>d</i> _a min	<i>d</i> _b max	<i>D</i> _a		<i>D</i> _b min	<i>S</i> _a min	<i>S</i> _b min
				min	max			
30	62	36	37	52	56	58	3	4.5
35	72	42	43	61	65	67	3	5.5
40	80	47	48	68	73	75	3	6.0
45	85	52	53	73	78	80	3	6.0

(continued)

<i>d</i>	322 series							
	<i>D</i>	<i>d_a</i>	<i>d_b</i>	<i>D_a</i>		<i>D_b</i>	<i>S_a</i>	<i>S_b</i>
		min	max	min	max			
50	90	57	58	78	83	85	3	6.0
55	100	64	63	87	91	95	4	6.0
60	110	69	69	95	101	104	4	6.0
65	120	74	75	104	111	115	4	6.0
70	125	79	80	108	116	119	4	6.5
75	130	84	85	115	121	125	4	6.5
80	140	90	90	122	130	134	5	7.5
85	150	95	96	130	140	143	5	8.5
90	160	100	101	138	150	153	5	8.5
95	170	107	107	145	158	162	5	8.5
100	180	112	113	154	168	171	5	10.0
105	190	117	119	161	178	180	5	10.0
110	200	122	125	170	188	191	6	10.0
120	215	132	135	181	203	205	7	11.5
130	230	144	144	193	216	220	7	14.0
140	250	154	157	210	236	239	8	14.0
150	270	164	170	226	256	256	8	17.0
160	290	174	181	242	276	276	10	17.0
170	310	188	194	259	292	296	10	20.0
180	320	198	202	267	302	305	10	20.0
190	340	208	215	286	322	325	10	22.0
200	360	218	222	302	342	342	11	22.0

Table 3 – 79 23 series tapered roller bearing installation dimension

(unit: mm)

<i>d</i>	323 series							
	<i>D</i>	<i>d_a</i>	<i>d_b</i>	<i>D_a</i>		<i>D_b</i>	<i>S_a</i>	<i>S_b</i>
		min	max	min	max			
17	17	23	24	39	41	43	3	4.5
20	52	27	27	43	45	47	3	4.5
25	62	32	33	52	55	57	3	5.5
30	72	37	39	59	65	66	4	6.0
35	80	44	44	66	71	74	4	8.0
40	90	49	50	73	81	82	4	8.5

(continued)

d	323 series							
	D	d_a	d_b	D_a		D_b	S_a	S_b
		min	max	min	max		min	min
45	100	54	56	82	91	93	4	8.5
50	110	60	62	90	100	102	5	9.5
55	120	65	68	99	110	111	5	10.5
60	130	72	73	107	118	121	6	11.5
65	140	77	80	117	128	131	6	12.0
70	150	82	86	125	138	140	6	12.0
75	160	87	91	133	148	150	7	13.0
80	170	92	98	142	158	160	7	13.5
85	180	99	103	150	166	168	8	14.5
90	190	104	108	157	176	178	8	14.5
95	200	109	114	166	186	187	8	16.5
100	215	114	123	177	201	201	8	17.5
105	225	119	128	185	211	210	8	18.5
110	240	124	137	198	226	223	9	19.5
120	260	134	148	213	246	241	9	21.5

3. 7. 7 Installation Dimensions of Thrust Ball Bearings

Figure 3 – 31 shows the thrust ball bearing installation dimension. During installing the thrust ball bearing, the shaft shoulder diameter (d_a) and the housing shoulder diameter (D_a) should conform to the values listed in Table 3 – 80.

Table 3 – 80 Thrust ball bearing installation dimension (unit: mm)

d	511 series			512 series			513 series			514 series		
	D	d_a	D_a	D	d_a	D_a	D	d_a	D_a	D	d_a	D_a
		min	max		min	max		min	max		min	max
10	24	18	16	26	20	16	—	—	—	—	—	—
12	26	20	18	28	22	18	—	—	—	—	—	—
15	28	23	20	32	25	22	—	—	—	—	—	—
17	30	25	22	35	28	24	—	—	—	—	—	—
20	35	29	26	40	32	28	—	—	—	—	—	—
25	42	35	32	47	38	34	52	41	36	60	46	39

(continued)

d	511 series			512 series			513 series			514 series		
	D	d_a min	D_a max	D	d_a min	D_a max	D	d_a min	D_a max	D	d_a min	D_a max
30	47	40	37	52	43	39	60	48	42	70	54	46
35	52	45	42	62	51	46	68	55	48	80	62	53
40	60	52	48	68	57	51	78	63	55	90	70	60
45	65	57	53	73	62	56	85	69	61	100	78	67
50	70	62	58	78	67	61	95	77	68	110	86	74
55	78	69	64	90	76	69	105	85	75	120	94	81
60	85	75	70	95	81	71	110	90	80	130	102	88
65	90	80	75	100	86	79	115	95	85	140	110	95
70	95	85	80	105	91	84	125	103	92	150	118	102
75	100	90	85	110	96	89	135	111	99	160	125	110
80	105	95	90	115	101	94	100	116	101	170	133	117
85	110	100	95	125	109	101	150	121	111	180	141	124
90	120	108	102	135	117	108	155	129	116	190	149	131
100	135	121	114	150	130	120	170	142	128	210	165	145
110	145	131	124	160	140	130	190	158	142	230	181	159
120	155	141	134	170	150	140	210	173	157	250	196	174
130	170	154	146	180	166	154	225	186	169	270	212	188
140	180	164	156	200	176	164	240	199	181	280	222	198
150	190	174	166	215	189	176	250	209	191	300	238	212
160	200	184	176	225	199	186	270	225	205	—	—	—
170	215	197	188	240	212	196	280	235	215	—	—	—
180	225	207	198	250	222	208	300	251	229	—	—	—
190	240	220	210	270	238	222	320	266	244	—	—	—
200	250	230	220	280	248	232	340	282	258	—	—	—
220	270	250	240	300	268	252	—	—	—	—	—	—
240	300	276	264	340	299	281	—	—	—	—	—	—
260	320	296	284	360	319	301	—	—	—	—	—	—
280	350	322	308	380	339	321	—	—	—	—	—	—
300	380	348	332	400	371	349	—	—	—	—	—	—
320	400	368	352	440	391	369	—	—	—	—	—	—
340	420	388	372	460	411	389	—	—	—	—	—	—
360	440	408	392	500	442	416	—	—	—	—	—	—

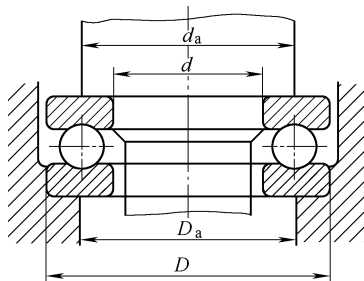


Figure 3-31 Thrust ball bearing installation dimension

3. 7. 8 Installation Dimensions of Thrust Self-Aligning Roller Bearings

Figure 3-32 shows the thrust self-aligning roller bearing installation dimension. During installing the thrust self-aligning roller bearing, the shaft shoulder diameter (d_a) and the housing shoulder diameter (D_a) should conform to the values listed in Table 3-81.

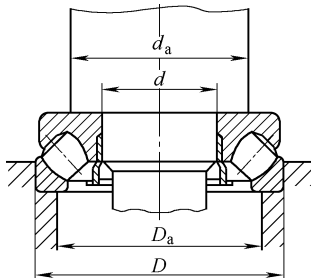


Figure 3-32 Thrust self-aligning roller bearing installation dimension

Table 3-81 Thrust self-aligning roller bearing installation dimension

(unit: mm)

d	292 series			293 series			294 series		
	D	d_a min	D_a max	D	d_a min	D_a max	D	d_a min	D_a max
60	—	—	—	—	—	—	130	90	107
65	—	—	—	—	—	—	140	100	115
70	—	—	—	—	—	—	150	105	124
75	—	—	—	—	—	—	160	115	132
80	—	—	—	—	—	—	170	120	141

(continued)

d	292 series			293 series			294 series		
	D	d_a min	D_a max	D	d_a min	D_a max	D	d_a min	D_a max
85	—	—	—	150	115	129	180	130	150
90	—	—	—	155	118	135	190	135	158
100	—	—	—	170	132	148	210	150	175
110	—	—	—	190	145	165	230	165	192
120	—	—	—	210	160	182	250	180	210
130	—	—	—	225	170	195	270	195	227
140	—	—	—	240	185	208	280	205	237
150	—	—	—	250	195	220	300	220	253
160	—	—	—	270	210	236	320	230	271
170	—	—	—	280	220	247	340	245	288
180	—	—	—	300	235	263	360	260	305
190	—	—	—	320	250	281	380	275	322
200	280	235	258	340	263	298	400	290	338
220	300	260	277	360	285	316	420	310	360
240	340	285	311	380	300	337	440	330	381
260	380	305	331	420	330	372	480	360	412
280	380	325	331	440	350	394	520	390	446
300	420	355	386	480	380	429	540	410	471
320	440	375	406	500	400	449	580	435	507
340	460	395	427	540	430	484	620	465	541
360	500	420	461	560	450	504	640	485	560
380	520	440	480	600	480	538	670	510	587
400	540	460	500	620	500	557	716	540	622
420	580	490	534	650	525	585	730	560	643
440	600	510	554	680	548	614	780	595	684
460	620	530	575	710	575	638	800	615	704
480	650	555	603	730	593	660	850	645	744
500	670	575	622	750	615	683	870	670	765
530	710	611	661	800	650	724	920	700	810
560	750	645	697	850	691	770	980	750	860
600	800	690	744	900	735	845	1030	800	900

The values listed in this Table are suitable for the light and medium loads, while increasing the load, it should increase the d_a and reduce the D_a .

3.7.9 Installation Dimensions of Radial Bearings with Adapter Sleeves

Figure 3 – 33 and Figure 3 – 34 show the thrust self-aligning roller bearing installation dimensions.

In order precisely to install the radial bearing with adapter sleeve on the shaft, usually, the inner ring must be supported against a shaft shoulder by a spacer, The spacer internal dimensions should ensure the bearing precise axial location, and easily mounting and dismounting for the adapter sleeve. Table 3 – 82 and Table 3 – 83 list the space internal dimensions.

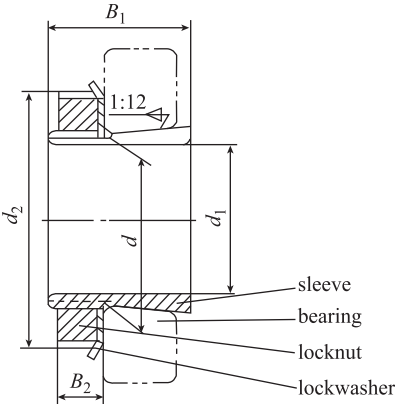


Figure 3 – 33 Installation dimension of the radial bearing with adapter sleeve (1)

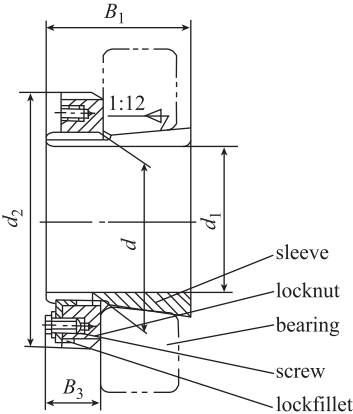


Figure 3 – 34 Installation dimension of the radial bearing with adapter sleeve (2)

Table 3 - 82 Installation dimension of the radial bearing with H2, H3, H23 series adapter sleeve (unit: mm)

d_1	d	d_2	B_1	B_2 max	G	adapter sleeve	sleeve	locknut	lockwasher
12	15	25	19	6	M15 * 1	H 202	A 202 X	KM 02	MB 02
		25	22	6	M15 * 1	H 302	A 302 X	KM 02	MB 02
		25	25	6	M15 * 1	H 2302	A 2302 X	KM 02	MB 02
14	17	28	20	6	M17 * 1	H 203	A 203 X	KM 03	MB 03
		28	24	6	M17 * 1	H 303	A 303 X	KM 03	MB 03
		28	27	6	M17 * 1	H 2303	A 2303 X	KM 03	MB 03
17	20	32	24	7	M20 * 1	H 204	A 204 X	KM 04	MB 04
		32	28	7	M20 * 1	H 304	A 304 X	KM 04	MB 04
		32	31	7	M20 * 1	H 2304	A 2304 X	KM 04	MB 04
20	25	38	26	8	M26 * 1.5	H 205	A 205 X	KM 05	MB 05
		38	29	8	M25 * 1.5	H 305	A 305 X	KM 05	MB 05
		38	35	8	M25 * 1.5	H 2305	A 2305 X	KM 05	MB 05
25	30	45	27	8	M30 * 1.5	H 206	A 206 X	KM 06	MB 06
		45	31	8	M30 * 1.5	H 306	A 306 X	KM 06	MB 06
		45	38	8	M30 * 1.5	H 2306	A 2306 X	KM 06	MB 06
30	35	52	29	9	M35 * 1.5	H 207	A 207 X	KM 07	MB 07
		52	35	9	M35 * 1.5	H 307	A 307 X	KM 07	MB 07
		52	43	9	M35 * 1.5	H 2307	A 2307 X	KM 07	MB 07

(continued)

d_1	d	d_2	B_1	B_2 max	G	adapter sleeve	sleeve	locknut	lockwasher
35	40	58	31	10	M40 * 1.5	H 208	A 208 X	KM 08	MB 08
		58	36	10	M40 * 1.5	H 308	A 308 X	KM 08	MB 08
		58	46	10	M40 * 1.5	H 2308	A 2308 X	KM 08	MB 08
40	45	65	33	11	M45 * 1.5	H 209	A 209 X	KM 09	MB 09
		65	39	11	M45 * 1.5	H 309	A 309 X	KM 09	MB 09
		65	50	11	M45 * 1.5	H 2309	A 2309 X	KM 09	MB 09
45	50	70	35	12	M50 * 1.5	H 210	A 210 X	KM 10	MB 10
		70	42	12	M50 * 1.5	H 310	A 310 X	KM 10	MB 10
		70	55	12	M50 * 1.5	H 2310	A 2310 X	KM 10	MB 10
50	55	75	37	12	M55 * 2	H 211	A 211 X	KM 11	MB 11
		75	45	12	M55 * 2	H 311	A 311 X	KM 11	MB 11
		75	59	12	M55 * 2	H 2311	A 2311 X	KM 11	MB 11
55	60	80	38	13	M60 * 2	H 212	A 212 X	KM 12	MB 12
		80	47	13	M60 * 2	H 312	A 312 X	KM 12	MB 12
		80	62	13	M60 * 2	H 2312	A 2312 X	KM 12	MB 12
60	65	85	40	14	M65 * 2	H 213	A 213 X	KM 13	MB 13
		85	50	14	M65 * 2	H 313	A 313 X	KM 13	MB 13
		85	65	14	M65 * 2	H 2313	A 2313 X	KM 13	MB 13

(continued)

d_1	d	d_2	B_1	B_2 max	G	adapter sleeve	sleeve	locknut	lockwasher
65	70	92	41	14	M70 * 2	H 214	A 214 X	KM 14	MB 14
		92	52	14	M70 * 2	H 314	A 314 X	KM 14	MB 14
		92	68	14	M70 * 2	H 2314	A 2314 X	KM 14	MB 14
	75	98	43	15	M75 * 2	H 215	A 215 X	KM 15	MB 15
		98	55	15	M75 * 2	H 315	A 315 X	KM 15	MB 15
		98	73	15	M75 * 2	H 2315	A 2315 X	KM 15	MB 15
	80	105	46	17	M80 * 2	H 216	A 216 X	KM 16	MB 16
		105	59	17	M80 * 2	H 316	A 316 X	KM 16	MB 16
		105	78	17	M80 * 2	H 2316	A 2316 X	KM 16	MB 16
75	85	110	50	18	M85 * 2	H 217	A 217 X	KM 17	MB 17
		110	63	18	M85 * 2	H 317	A 317 X	KM 17	MB 17
		110	82	18	M85 * 2	H 2317	A 2317 X	KM 17	MB 17
80	90	120	52	18	M90 * 2	H 218	A 218 X	KM 18	MB 18
		120	65	18	M90 * 2	H 318	A 318 X	KM 18	MB 18
		120	86	18	M90 * 2	H 2318	A 2318 X	KM 18	MB 18
85	95	125	55	19	M95 * 2	H 219	A 219 X	KM 19	MB 19
		125	68	19	M95 * 2	H 319	A 319 X	KM 19	MB 19
		125	90	19	M95 * 2	H 2319	A 2319 X	KM 19	MB 19

(continued)

d_1	d	d_2	B_1	B_2 max	G	adapter sleeve	sleeve	locknut	lockwasher
90	100	130	58	20	M100 * 2	H 220	A 220 X	KM 20	MB 20
		130	71	20	M100 * 2	H 320	A 320 X	KM 20	MB 20
		130	76	20	M100 * 2	H 3120	A 3120 X	KM 20	MB 20
		130	97	20	M100 * 2	H 2320	A 2320 X	KM 20	MB 20
95	105	140	60	20	M105 * 2	H 221	A 221 X	KM 21	MB 21
		140	74	20	M105 * 2	H 321	A 321 X	KM 21	MB 21
		140	80	20	M105 * 2	H 3121	A 3121 X	KM 21	MB 21
		140	101	20	M105 * 2	H 2321	A 2321 X	KM 21	MB 21
100	110	145	63	21	M110 * 2	H 222	A 222 X	KM 22	MB 22
		145	77	21	M110 * 2	H 322	A 322 X	KM 22	MB 22
		145	81	21	M110 * 2	H 3122	A 3122 X	KM 22	MB 22
		145	105	21	M110 * 2	H 2322	A 2322 X	KM 22	MB 22
110	120	145	60	22	M120 * 2	H 3924	A 3924 X	KML 24	MBL 24
		145	72	22	M120 * 2	H 3024	A 3024 X	KML 24	MBL 24
		155	88	22	M120 * 2	H 3124	A 3124 X	KM 24	MB 24
		155	112	22	M120 * 2	H 2324	A 2324 X	KM 24	MB 24
115	130	155	65	23	M130 * 2	H 3926	A 3926 X	KML 26	MBL 26
		155	80	23	M130 * 2	H 3026	A 3026 X	KML 26	MBL 26
		165	92	23	M130 * 2	H 3126	A 3126 X	KM 26	MB 26

(continued)

d_1	d	d_2	B_1	B_2 max	G	adapter sleeve	sleeve	locknut	lockwasher
125	140	165	121	23	M130 * 2	H 2326	A 2326 X	KM 26	MB 26
		165	66	24	M140 * 2	H 3928	A 3928 X	KML 28	MBL 28
		165	82	24	M140 * 2	H 3028	A 3028 X	KML 28	MBL 28
		180	97	24	M140 * 2	H 3128	A 3128 X	KM 28	MB 28
		180	131	24	M140 * 2	H 2328	A 2328 X	KM 28	MB 28
135	150	180	76	26	M150 * 2	H 3930	A 3930 X	KML 30	MBL 30
		180	87	26	M150 * 2	H 3030	A 3030 X	KML 30	MBL 30
		195	111	26	M150 * 2	H 3130	A 3130 X	KM 30	MB 30
		195	139	26	M150 * 2	H 2330	A 2330 X	KM 30	MB 30
140	160	190	78	28	M160 * 3	H 3932	A 3932 X	KML 32	MBL 32
		190	93	28	M160 * 3	H 3032	A 3032 X	KML 32	MBL 32
		210	119	28	M160 * 3	H 3132	A 3132 X	KM 32	MB 32
		210	147	28	M160 * 3	H 2332	A 2332 X	KM 32	MB 32
150	170	200	79	29	M170 * 3	H 3934	A 3934 X	KML 34	MBL 34
		200	101	29	M170 * 3	H 3034	A 3034 X	KML 34	MBL 34
		220	122	29	M170 * 3	H 3134	A 3134 X	KM 34	MB 34
		220	154	29	M170 * 3	H 2334	A 2334 X	KM 34	MB 34
160	180	210	87	30	M180 * 3	H 3936	A 3936 X	KML 36	MBL 36

(continued)

d_1	d	d_2	B_1	B_2 max	G	adapter sleeve	sleeve	locknut	lockwasher
170	190	210	109	30	M180 * 3	H 3036	A 3036 X	KML 36	MBL 36
		230	131	30	M180 * 3	H 3136	A 3136 X	KM 36	MB 36
		230	161	30	M180 * 3	H 2336	A 2336 X	KM 36	MB 36
		220	89	31	M190 * 3	H 3938	A 3938 X	KML 38	MBL 38
		220	112	31	M190 * 3	H 3038	A 3038 X	KML 38	MBL 38
		240	141	31	M190 * 3	H 3138	A 3138 X	KM 38	MB 38
		240	169	31	M190 * 3	H 2338	A 2338 X	KM 38	MB 38
180	200	240	98	32	M200 * 3	H 3940	A 3940 X	KML 40	MBL 40
		240	120	32	M200 * 3	H 3040	A 3040 X	KML 40	MBL 40
		250	150	32	M200 * 3	H 3140	A 3140 X	KM 40	MB 40
		250	176	32	M200 * 3	H 2340	A 2340 X	KM 40	MB 40
200	220	280	161	35	Tr220 * 4	H 3144	A 3144	HM 44	MB 44
		280	186	35	Tr220 * 4	H 2344	A 2344	HM 44	MB 44
220	240	300	172	37	Tr240 * 4	H 3148	A 3148	HM 48	MB 48
		300	199	37	Tr240 * 4	H 2348	A 2348	HM 48	MB 48
240	260	330	190	39	Tr260 * 4	H 3152	A 3152	HM 52	MB 52
		330	211	39	Tr260 * 4	H 2352	A 2352	HM 52	MB 52
260	280	350	195	41	Tr280 * 4	H 3156	A 3156	HM 56	MB 56
		350	224	41	Tr280 * 4	H 2356	A 2356	HM 56	MB 56

Table 3 – 83 Installation dimension of the radial bearing with H30, H31, H32, H39 series adapter sleeve (unit: mm)

d_1	d	d_2	B_1	B_3	G	adapter sleeve	sleeve	locknut	lockfillet
200	220	260	96	41	Tr240 * 4	H 3944	A 3944	HML 44	MSL 44
		260	126	41	Tr240 * 4	H 3044	A 3044	HML 44	MSL 44
220	240	290	101	46	Tr220 * 4	H 3948	A 3948	HML 48	MSL 48
		290	133	46	Tr220 * 4	H 3048	A 3048	HML 48	MSL 48
240	260	310	116	46	Tr260 * 4	H 3952	A 3952	HML 48	MSL 48
		310	145	46	Tr260 * 4	H 3052	A 3052	HML 48	MSL 48
260	280	330	121	50	Tr280 * 4	H 3956	A 3956	HML 56	MSL 56
		330	152	50	Tr280 * 4	H 3056	A 3056	HML 56	MSL 56
280	300	360	140	54	Tr300 * 4	H 3960	A 3960	HML 60	MSL 60
		360	168	54	Tr300 * 4	H 3060	A 3060	HML 60	MSL 60
		380	208	53	Tr300 * 4	H 3160	A 3160	HM 60	MS 60
		380	240	53	Tr300 * 4	H 3260	A 3260	HM 60	MS 60
300	320	380	140	55	Tr320 * 5	H 3964	A 3964	HML 64	MSL 64
		380	171	55	Tr320 * 5	H 3064	A 3064	HML 64	MSL 64
		400	226	56	Tr320 * 5	H 3164	A 3164	HM 64	MS 64
		400	258	56	Tr320 * 5	H 3264	A 3264	HM 64	MS 64

(continued)

d_1	d	d_2	B_1	B_3	G	adapter sleeve	sleeve	locknut	lockfillet
320	340	400	144	58	Tr340 * 5	H 3968	A 3968	HML 68	MSL 64
		400	187	58	Tr340 * 5	H 3068	A 3068	HML 68	MSL 64
		440	254	72	Tr340 * 5	H 3168	A 3168	HM 68	MS 68
		440	288	72	Tr340 * 5	H 3268	A 3268	HM 68	MS 68
340	360	420	144	58	Tr360 * 5	H 3972	A 3972	HML 72	MSL 72
		420	188	58	Tr360 * 5	H 3072	A 3072	HML 72	MSL 72
		460	259	75	Tr360 * 5	H 3172	A 3172	HM 72	MS 68
		460	299	75	Tr360 * 5	H 3272	A 3272	HM 72	MS 68
360	380	450	164	62	Tr380 * 5	H 3976	A 3976	HML 76	MSL 76
		450	193	62	Tr380 * 5	H 3076	A 3076	HML 76	MSL 76
		490	264	77	Tr380 * 5	H 3176	A 3176	HM 76	MS 76
		490	310	77	Tr380 * 5	H 3276	A 3276	HM 76	MS 76
380	400	470	168	66	Tr400 * 5	H 3980	A 3980	HML 80	MSL 80
		470	210	66	Tr400 * 5	H 3080	A 3080	HML 80	MSL 80
		520	272	82	Tr400 * 5	H 3180	A 3180	HM 80	MS 80
		520	328	82	Tr400 * 5	H 3280	A 3280	HM 80	MS 80
400	420	490	168	66	Tr420 * 5	H 3984	A 3984	HML 84	MSL 84
		490	212	66	Tr420 * 5	H 3084	A 3084	HML 84	MSL 84

(continued)

d_1	d	d_2	B_1	B_3	G	adapter sleeve	sleeve	locknut	lockfillet
410	440	540	304	90	Tr420 * 5	H 3184	A 3184	HM 84	MS 80
		540	352	90	Tr420 * 5	H 3284	A 3284	HM 84	MS 80
		520	189	77	Tr440 * 5	H 3988	A 3988	HML 88	MSL 88
		520	228	77	Tr440 * 5	H 3088	A 3088	HML 88	MSL 88
		560	307	90	Tr440 * 5	H 3188	A 3188	HM 88	MS 88
		560	361	90	Tr440 * 5	H 3288	A 3288	HM 88	MS 88
430	460	540	189	77	Tr460 * 5	H 3992	A 3992	HML 92	MSL 88
		540	234	77	Tr460 * 5	H 3092	A 3092	HML 92	MSL 88
		580	326	95	Tr460 * 5	H 3192	A 3192	HM 92	MS 88
		580	382	95	Tr460 * 5	H 3292	A 3292	HM 92	MS 88
450	480	560	200	77	Tr480 * 5	H 3996	A 3996	HML 96	MSL 96
		560	237	77	Tr480 * 5	H 3096	A 3096	HML 96	MSL 96
		620	335	95	Tr480 * 5	H 3196	A 3196	HM 96	MS 96
		620	397	95	Tr480 * 5	H 3296	A 3296	HM 96	MS 96
470	500	580	208	85	Tr500 * 5	H 39/500	A 39/500	HML /500	MSL 96
		580	247	85	Tr500 * 5	H 30/500	A 30/500	HML /500	MSL 96
		630	356	100	Tr500 * 5	H 31/500	A 31/500	HM /500	MS /500
		630	428	100	Tr500 * 5	H 32/500	A 32/500	HM /500	MS /500

(continued)

d_1	d	d_2	B_1	B_3	G	adapter sleeve	sleeve	locknut	lockfillet
500	530	630	216	90	Tr530 * 6	H 39/530	A 39/530	HML/530	MSL/530
		630	265	90	Tr530 * 6	H 30/530	A 30/530	HML/530	MSL/530
		670	364	105	Tr530 * 6	H 31/530	A 31/530	HM/530	MS/530
		670	447	105	Tr530 * 6	H 32/530	A 32/530	HM/530	MS/530
530	560	650	227	97	Tr560 * 6	H 39/560	A 39/560	HML/560	MSL/560
		650	282	97	Tr560 * 6	H 30/560	A 30/560	HML/560	MSL/560
		710	377	110	Tr560 * 6	H 31/560	A 31/560	HM/560	MS/560
		710	462	110	Tr560 * 6	H 32/560	A 32/560	HM/560	MS/560
560	600	700	239	97	Tr600 * 6	H 39/600	A 39/600	HML/600	MSL/600
		700	289	97	Tr600 * 6	H 30/600	A 30/600	HML/600	MSL/600
		750	399	110	Tr600 * 6	H 31/600	A 31/600	HM/600	MS/600
		750	487	110	Tr600 * 6	H 32/600	A 32/600	HM/600	MS/600
600	630	730	254	97	Tr630 * 6	H 39/630	A 39/630	HML/630	MSL/630
		730	301	92	Tr630 * 6	H 30/630	A 30/630	HML/630	MSL/630
		800	424	120	Tr630 * 6	H 31/630	A 31/630	HM/630	MS/630
		800	521	120	Tr630 * 6	H 32/630	A 32/630	HM/630	MS/630

(continued)

d_1	d	d_2	B_1	B_3	G	adapter sleeve	sleeve	locknut	lockfillet
630	670	780	264	102	Tr670 * 6	H 39/670	A 39/670	HML/670	MSL/670
		780	324	102	Tr670 * 6	H 30/670	A 30/670	HML/670	MSL/670
		850	456	131	Tr670 * 6	H 31/670	A 31/670	HM/670	MS/670
		850	558	131	Tr670 * 6	H 32/670	A 32/670	HM/670	MS/670
670	710	830	286	112	Tr710 * 6	H 39/710	A 39/710	HML/710	MSL/710
		830	342	112	Tr710 * 6	H 30/710	A 30/710	HML/710	MSL/710
		900	467	135	Tr710 * 6	H 31/710	A 31/710	HM/710	MS/710
		900	572	135	Tr710 * 6	H 32/710	A 32/710	HM/710	MS/710
710	750	870	291	112	Tr750 * 6	H 39/750	A 39/750	HML/750	MSL/750
		870	356	112	Tr750 * 6	H 30/750	A 30/750	HML/750	MSL/750
		950	493	141	Tr750 * 6	H 31/750	A 31/750	HM/750	MS/750
		950	603	141	Tr750 * 6	H 32/750	A 32/750	HM/750	MS/750
750	800	920	303	112	Tr800 * 6	H 39/800	A 39/800	HML/800	MSL/750
		920	366	112	Tr800 * 6	H 30/800	A 30/800	HML/800	MSL/750
		1000	505	141	Tr800 * 6	H 31/800	A 31/800	HM/800	MS/750
		1000	618	141	Tr800 * 6	H 32/800	A 32/800	HM/800	MS/800

(continued)

d_1	d	d_2	B_1	B_3	G	adapter sleeve	sleeve	locknut	lockfillet
800	850	980	308	115	Tr850 * 6	H 39/850	A 39/850	HML/850	MSL/850
		980	380	115	Tr850 * 6	H 30/850	A 30/850	HML/850	MSL/850
		1060	536	147	Tr850 * 6	H 31/850	A 31/850	HM/850	MS/850
		1060	651	147	Tr850 * 6	H 32/850	A 32/850	HM/850	MS/850
850	900	1030	326	125	Tr900 * 6	H 39/900	A 39/900	HML/900	MSL/850
		1030	400	125	Tr900 * 6	H 30/900	A 30/900	HML/900	MSL/850
		1120	557	154	Tr900 * 6	H 31/900	A 31/900	HM/900	MS/900
		1120	660	154	Tr900 * 6	H 32/900	A 32/900	HM/900	MS/900
900	950	1080	344	125	Tr950 * 6	H 39/950	A 39/950	HML/950	MSL/950
		1080	420	125	Tr950 * 6	H 30/950	A 30/950	HML/950	MSL/950
		1170	583	154	Tr950 * 6	H 31/950	A 31/950	HM/950	MS/950
		1170	675	154	Tr950 * 6	H 32/950	A 32/950	HM/950	MS/950
950	1000	1140	358	125	Tr1000 * 6	H 39/1000	A 39/1000	HML/1000	MSL/1000
		1140	430	125	Tr1000 * 6	H 30/1000	A 30/1000	HML/1000	MSL/1000
		1240	609	154	Tr1000 * 6	H 31/1000	A 31/1000	HM/1000	MS/1000
		1240	707	154	Tr1000 * 6	H 32/1000	A 32/1000	HM/1000	MS/1000
1000	1060	1200	372	125	Tr1060 * 6	H 39/1060	A 39/1060	HML/1060	MSL/1000
		1200	447	125	Tr1060 * 6	H 30/1060	A 30/1060	HML/1060	MSL/1000
		1300	622	154	Tr1060 * 6	H 31/1060	A 31/1060	HM/1060	MS/10000

3.8 Calculation Example of Rolling Bearing Support Structure Design

The main contents of rolling bearing support structure design calculation are to:

- 1) decide the support combination form and the bearing arrangement.
- 2) conduct bearing selecting calculation.

However, the shaft system is an entirety that includes the transmission component, the shaft and the bearings, so, the designer has to know every part in the shaft system, including their carrying load, their strength and their designing calculation. The fundamental factors are the transmission power and the rotation speed condition that are the basis of design and calculation. But the shaft diameter on the bearing seat is the base of selecting bearing calculation. Therefore, before the bearing selecting calculation, the following calculations must have been finished:

- 1) Every shaft power and running factor calculations.
- 2) Gear transmission factor and load calculations.
- 3) Original deciding support span, calculating the acting forces on every support.
- 4) Shaft strength calculation for deciding the shaft diameter on the bearing seat to meet the requirement of shaft strength.

Generally, during the bearing selection calculation process, it always is the way to use two plans for analyzing and comparing, then to select the optimum bearing model.

At last, it should also be considered to select the bearing accuracy, the matching fits and the bearing clearance, the lubrication and sealing device.

In order to introduce the bearing selecting calculation process, the two stage taper and cylindrical gear box can be as an example. Figure 3-35 is its driving sketch, and there are three shafts in it.

the transmission power N is 10 kW, the input speed n_1 is 960r/min, the output speed n_{III} is 80r/min. The shaft III is subjecting to the external load (belt pulling force), $Q = 6\text{kN}$ or $Q = 9\text{kN}$. The support design

calculation process is as follows:

(1) The support construction and the bearing arrangement

The input shaft pinion is an overhung arrangement. In order to keep the correct engagement of the gear transmission and to let the shaft system having good rigidity, the span distance between the support *A* and the support *B* should be two or three times of the pinion overhung distance, and the cylindrical roller bearing should be selected. The support structure should be the combination structure of one fixed support and one floating support that is favorable for the adjustment of the gear engaging position. The locating supporter position should also be arranged so, that it is favorable for the adjustment of the bearing axial position and its floating clearance. It is recommended to select a pair of tapered roller bearings in face to face arrangement for the fixed support, and for the floating support to select the cylindrical roller bearing that has the large load carrying capacity and good rigidity.

The middle shaft is a combination structure of two supports limiting single direction position on the two ends of the shaft. Each of the two supports uses an identical tapered roller bearing that is face to face arrangement. This support structure is simple and favorable for the adjustment of the large tapered roller bearing axial position and the adjustment of the bearing floating clearance. The axial position can be adjusted to ensure the coincident of the two taper apexes when the larger gear and the smaller gear are engaged.

The output shaft is a combination structure of two supports limiting single direction position on the two ends of the shaft. The two supports can not only select two identical tapered roller bearings that are face to face arrangement, but also can select two self-aligning roller bearings consisting of two supports limiting single direction position on the two ends of the shaft. The self-aligning roller bearing can subject to heavy load and having good aligning performance, suitable for the applications where the power is large and the external pulling force Q is heavy.

Figure 3 – 36 is the three shaft support structures and the bearing arrangements on every shaft. The support positions are expressed by *A*, *B*,

C , D , E and F respectively.

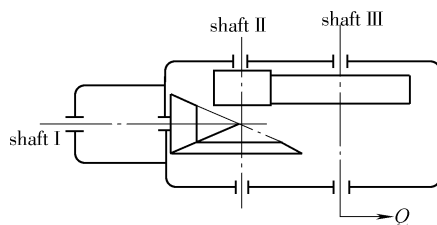


Figure 3-35 Diagram of two stage taper cylindrical gear transmission

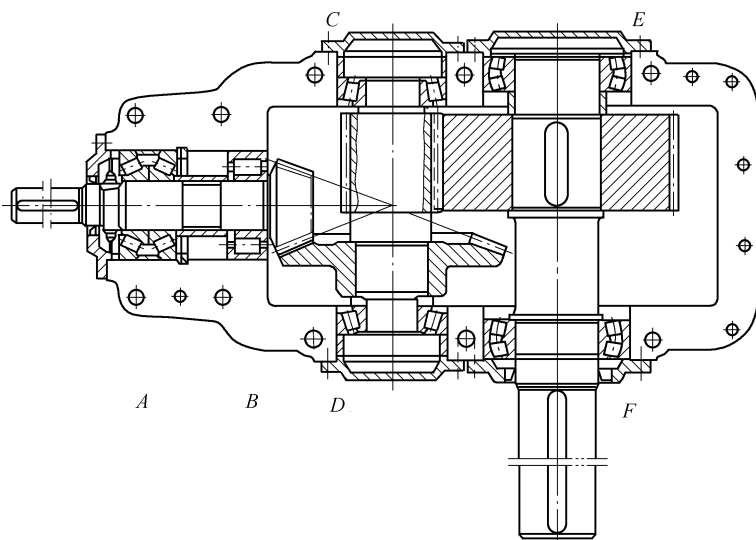


Figure 3-36 Bearing arrangement for two stage taper-cylindrical gear transmission

(2) Every shaft rotation and power factors

Table 3-84 lists the every shaft rotation and power factors.

Table 3-84 Every shaft rotation and power factors

shaft	Rotation speed $n/(\text{r/min})$	Transmission ratio i	Rotation moment $T/\text{N} \cdot \text{m}$
shaft I	960	3.17 3.80	99.5
shaft II	302.84		315.35
shaft III	79.6		1198.32

(3) Gear transmission factor and acting force calculation

Table 3 – 85 lists the gear materials, the heat treatments and calculated factors, Table 3 – 86 lists the circumference forces, the radial forces and the axial forces of every gear.

(4) Assume the support spans, the relative positions of force acting point and the acting forces on every shaft.

Figure 3 – 37 is the diagram of the every shaft support span and the force acting point position.

Table 3 – 87 lists the calculation result of every shaft acting force and the support acting force.

(5) Calculate the shaft strength and determine the shaft diameter on the bearing position.

The material of the three shafts is all the No.45 steel, through harmonious heat treatment, the hardness is 250 HBS.

Figure 3 – 38 is the diagram of the bending moment, rotation torque of the shaft I.

Figure 3 – 39 is the diagram of the bending moment, rotation torque of the shaft II.

Figure 3 – 40 is the diagram of the bending moment, rotation torque of the shaft III.

Table 3 – 85 Gear transmission factor calculation

Item	Spur taper gear		Bevel gear	
	small gear	large gear	small	large gear
material, heat treatment	No. 45 steel harmonious 300 HBS	No. 45 steel harmonious 250 HBS	No. 45 steel harmonious 300 HBS	No. 45 steel harmonious 250 HBS
gear	$z_1 = 23$	$z_2 = 73$	$z_3 = 20$	$z_4 = 76$
modulus	$m = 4$		$m = 4$	
diameter	$d_1 = 92$	$d_2 = 292$	$d_3 = 81.25$	$d_4 = 308.75$
angle	$\delta_1 = 17.51^\circ$	$\delta_2 = 72.49^\circ$	$\beta_1 = 10.06^\circ$ (right hand)	$\beta_2 = 10.06^\circ$ (left hand)
tooth wide/mm	$b_1 = 45$	$b_2 = 45$	$b_1 = 100$	$b_2 = 95$
	taper axes distance $R = 151.7\text{mm}$		center distance $a = 195\text{mm}$	

Table 3 – 86 Gear force calculation

Item	Spur taper gear		Bevel gear	
	small gear	large gear	small	large gear
calculation equation	circumference force $F_{r1} = -F_{a2} = F_{t1} \tan \alpha \cos \delta_1$		$F_{r3} = -F_{r4} = F_{t3} \tan \alpha / \cos \beta$	
	radial force $F_{a1} = -F_{r2} = F_{t1} \tan \alpha \sin \delta_1$		$F_{a3} = -F_{a4} = F_{t3} \tan \beta$	
	axial force $F_{t1} = -F_{t2} = \frac{2T_1}{d_{m1}}$		$F_{t3} = -F_{t4} = \frac{2T_2}{d_3}$	
force value /kN	$F_{t1} = 2.54$ $F_{t2} = 2.54$		$F_{t3} = 7.76$ $F_{t4} = 7.76$	
	$F_{r1} = 0.884$ $F_{r2} = 0.277$		$F_{r3} = 1.395$ $F_{r4} = 1.395$	
	$F_{a1} = 0.277$ $F_{a2} = 0.884$		$F_{a3} = 1.374$ $F_{a4} = 1.374$	

Table 3 – 87 Every shaft acting force and support acting force

(unit: kN)

Shaft	Gear circumference force F_t	Gear radial force F_r	Gear axial force F_a	Pulling force Q	Supporter radial force
I	F_{t1} 2.54	F_{r1} 0.884	F_{a1} 0.277		F_{rA} 0.875
					F_{rB} 3.563
II	F_{t2} 2.54	F_{r2} 0.277	F_{a2} 0.884		F_{rC} 6.535
	F_{t3} 7.76	F_{r3} 1.395	F_{a3} 1.374		F_{rD} 3.984
III	F_{tr} 7.76	F_{r4} 1.395	F_{a4} 1.374	Q 6	F_{rE} 5.81
					F_{rF} 11.27
	F_{t4} 7.76	F_{r4} 1.395	F_{a4} 1.374	Q 9	F_{rE} 6.658
					F_{rF} 16.074

Table 3 – 88 Lists the first selecting every support bearing position shaft diameter meeting the requirement for strength.

Table 3 – 88 First select shaft diameters on support bearing positions

shaft	shaft diameter dimension/mm (danger section)	bearing position shaft diameter/mm
shaft I	on support A $d_A \geq 27$	$d_A = 35$
	on support B $d_B \geq 31.8$	$d_B = 35$
shaft II	on wheel 3 (closing to C) $d_{II3} \geq 47.9$	$d_C = 50$
	on support D $d_B \geq 43.5$	$d_D = 50$

(continued)

shaft	shaft diameter dimension/mm (danger section)	bearing position shaft diameter/mm
shaft III ($Q = 6 \text{ KN}$)	on wheel 4 (closing to E) $d_{\text{III}4} \geq 58.2$	$d_E = 65$
	on support F $d_F \geq 59.4$	$d_F = 65$
shaft III ($Q = 9 \text{ KN}$)	on wheel 4 (closing to E) $d_{\text{III}4} \geq 59.4$	$d_E = 70$
	on support F $d_F \geq 69.8$	$d_F = 70$

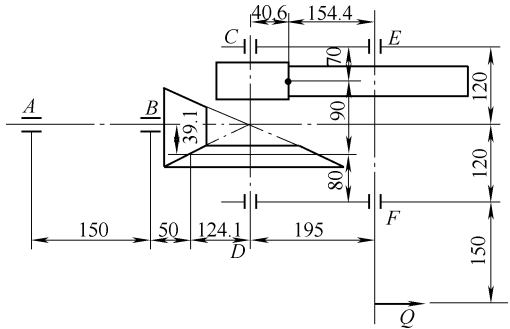


Figure 3-37 Diagram of the every shaft support span and the force acting point position

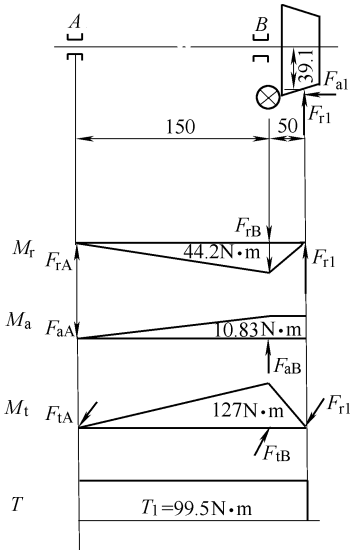


Figure 3-38 Diagram of the bending moment, rotation torque of the shaft I

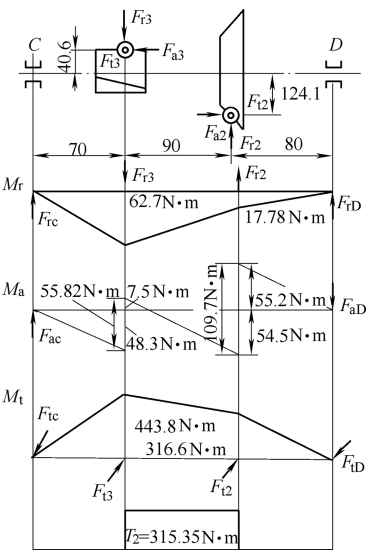


Figure 3-39 Diagram of the bending moment, rotation torque of the shaft II

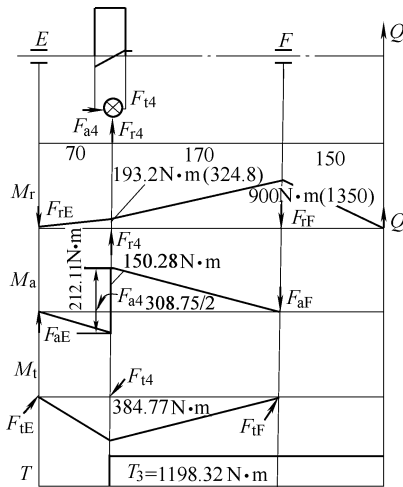


Figure 3-40 Diagram of the bending moment, rotation torque of the shaft III

(6) Bearing selecting calculation

The bearing selecting calculation is the bearing fatigue life calculation. It can be done by selecting the bearing model according to the required dynamic capacity C_j , but it also can be done first to select the bearing model and then to check if the selected bearing could meet the requirement according to the selected bearing dynamic load rating C . The former can ensure that the bearings subjecting to heavy load on every support would have the near same fatigue life. However, no matter which method to be used, the first thing is to select the bearing model in advance to get the bearing calculation factors.

When selecting the bearing model in advance, it should be considered that:

1) The bearing inner ring must meet the shaft strength requirement which should be decided according to the shaft strength. If the calculation result can not meet the requirement for the bearing load capacity, then it is recommended to select another bearing model which inner ring is larger.

2) Usually one shaft has two supports, when using the same type

bearings, the same model bearings should be used; when using different type bearings, it should be selected that the bearings have the same inner ring diameters and the same outer diameters.

3) Only to take calculation for the support that is subjected to heavier load.

Table 3 – 89 lists the first selecting bearing models and their calculation factors.

Table 3 – 90 lists the dynamic equivalent load P calculation. For the tapered roller bearing, the dynamic equivalent load calculation equation is as follows:

$$P = F_r \quad \text{for } F_a/F_r < e;$$
$$P = XF_r + YF_a \quad \text{for } F_a/F_r \geq e;$$

Table 3 – 89 Bearing model and calculation factors (selected in advance)

Item	Support A	Support B	Support C, D	Support E, F (Q = 6kN)	Support E, F (Q = 9kN)
bearing model	30207	NUP 207	30210	30213	30214
calculation factor	$C = 56.8\text{kN}$ $e = 0.37$ $Y = 1.6$	$C = 48.8\text{kN}$	$C = 76.8\text{kN}$ $e = 0.42$ $Y = 1.4$	$C = 125\text{kN}$ $e = 0.40$ $Y = 1.5$	$C = 138\text{kN}$ $e = 0.42$ $Y = 1.4$

Note: C ——load rating e ——judgement factor Y ——axial factor.

Table 3 – 90 Each bearing dynamic equivalent load P (kN)

Shaft	Bearing position	Radial force F_r	Interior axial force S	Shaft axial force F_A	Bearing axial force F_a	Equivalent load P
I	A	F_{rA} 0.875	S_{A1} 0.138	F_{a1} 0.277	F_{aA} 0.414	P_A 0.837
	B	F_{rB} 3.563	S_B 0	(directs to A)	F_{aB} 0	P_B 3.563
II	C	F_{rC} 6.535	S_C 2.334	F_{aII} 0.49	F_{aC} 2.334	P_C 6.535
	D	F_{rD} 3.984	S_D 1.423	(same direction as S_D)	F_{aD} 1.844	P_D 4.175
III $Q = 6\text{kN}$	E	F_{rE} 5.81	S_E 1.937	F_{aIII} 1.374	F_{aE} 2.383	P_E 5.917
	F	F_{rF} 11.27	S_F 3.757	(same as SE)	F_{aF} 3.757	P_F 11.27
III $Q = 9\text{kN}$	E	F_{rE} 6.658	S_E 2.378	F_{aIII} 1.374	F_{aE} 4.367	P_E 8.777
	F	F_{rF} 16.074	S_F 5.741	(same as SE)	F_{aF} 5.741	P_F 16.074

The calculation of bearing axial force should consider the interior axial force S caused from the radial force, $S = F_r/2Y$.

The gear box usage life should be 20000h, the load factor f_F is 1.2, the dynamic load rating C_j equation is as:

$$C_j = f_h * f_F / f_n * P$$

Table 3 – 91 lists the calculation result of each bearing.

Table 3 – 91 Dynamic load rating C_j calculation result of each bearing

Shaft	Bearing position	$n/$ (r/min)	f_n	f_L	Dynamic equivalent load P/kN	Dynamic load rating C_j/kN	Selected bearing C/kN
shaft I	A	960	0.365	3.02	P_A 0.837	C_{jA} 8.340	30207 $C = 56.8$
	B	960	0.365	3.02	P_B 3.563	C_{jB} 35.376	NUP207 $C = 48.8$
shaft II	C	303	0.516	3.02	P_C 6.535	C_{jC} 45.897	30210 $C = 76.8$
	D	303	0.516	3.02	P_D 4.175	C_{jD} 29.322	
shaft III $Q = 6\text{kN}$	E	80	0.769	3.02	P_E 5.917	C_{jE} 27.815	30213 $C = 125$
	F	80	0.769	0.32	P_F 11.27	C_{jF} 53.1	
shaft III $Q = 9\text{kN}$	E	80	0.769	3.02	P_E 8.777	C_{jE} 41.363	30214 $C = 138$
	F	80	0.769	3.02	P_F 16.074	C_{jF} 75.75	

It is not difficult to see that the bearing model of the support A on the shaft I is determined according to the inner ring diameter and the outer ring diameter of the bearing on support B; for the bearings of the supports E and F on the shaft III, all are the tapered roller bearings, their life calculation are also carried out according to the factors of tapered roller bearing.

Table 3 – 92 lists the bearing models of every support and their factors.

Table 3 – 92 Bearing models and factors

Bearing model	Dimensions	Dynamic load rating C/kN	Speed limit (grease) $n/$ (r/min)	Support
tapered roller bearing 30207	35 * 72 * 17	56.8	5300	A
cylindrical roller bearing NUP 207	35 * 72 * 17	48.8	7500	B
tapered roller bearing 30210	50 * 90 * 20	47.53	4300	C, D
tapered roller bearing 30213	65 * 120 * 23	64	3200	E, F
tapered roller bearing 30214	70 * 125 * 24	76.24	3000	E, F

The self-aligning roller bearing is suitable for carrying heavy radial load, but only can carry a little axial load. The supports E and F on shaft III are selected to use the self-aligning roller bearings, in this case, the dynamic equivalent load P_r and the required dynamic load rating C_j must be calculated. Table 3-93 lists the selected self-aligning roller bearing models and their factors.

Table 3-93 The selected self-aligning roller bearing models and their factors

Bearing model	Dimensions $d * D * B$	Dynamic load rating C/kN	Speed limit $/(r/\text{min})$	Calculation factor		
				e	Y_1	Y_2
22213 ($Q = 6\text{kN}$)	$65 * 120 * 31$	155	3200 3600 (grease) (oil)	0.25	2.7	4.0
22214 ($Q = 9\text{kN}$)	$70 * 125 * 31$	92.2	3000 3400 (grease) (oil)	0.24	2.9	4.3

The calculating equations of the dynamic equivalent load P_r and the required dynamic load rating C_j for the self-aligning roller bearing are as follows:

$$P_r = F_r + Y_1 * F_a \quad \text{for } F_a/F_r \leq e;$$
$$P_r = 0.67 F_r + Y_2 * F_a \quad \text{for } F_a/F_r > e.$$
$$C_j = (f_h * f_r/f_n) * P_r.$$

The values of f_h , f_r and f_n are taken as:

$$f_h = 3.02 \text{ (20000h)}; f_r = 1.2 \text{ and } f_n = 0.769 \text{ (} n_{III} = 80\text{r/min)}.$$

Table 3-94 lists the calculation results of the dynamic equivalent load P_r and the required dynamic load rating C_j of the supports E and F .

Table 3-94 The calculation results of the dynamic equivalent load P_r and the required dynamic load rating C_j of the supports E and F

Load	Support position	Bearing model	Radial load F_r/kN	Axial load F_a/kN	F_a/F_r	P/kN	C_j/kN	C/kN
$Q = 6\text{kN}$	E	22213	5.81	0		5.81		86.8
	F	22213	11.27	1.374	$0.122 < e$	14.57	68.66	86.8
$Q = 9\text{kN}$	E	22214	6.658	0		6.658		92.2
	F	22214	16.074	1.374	$0.085 < e$	19.37	91.29	92.2

Note: P ——dynamic equivalent load; C_j ——required dynamic load rating;
 C ——bearing dynamic load rating.

In this example, the tapered roller bearing and the self-aligning roller bearing have the same inner ring bore diameter and the same outer ring outer diameter, and their carrying load capabilities are quite equal to each other. This is because the self-aligning roller bearing has little axial load capability but has good self-aligning capability that is favorable for the wheel gear engaging.

(7) Selecting bearing accuracy, fits and clearance

1) Bearing Accuracy

The gearbox is an ordinary machinery, so, the 0 grade accuracy can meet the requirement for the support. Therefore, the 0 grade accuracy is selected for all bearings.

2) Fits

Many factors should be taken into account for the bearing inner ring fit with shaft and the bearing outer ring fit with the housing, such as the accuracy, the load condition, the clearance value and the clearance adjustment, the bearing dimension etc. Generally, the bearing fit selecting should take the recommended value according to the China National Standard GB/T 275—1993. Every bearing inner ring is subjected to the cycling load, every outer ring is subjected to the local load. GB/T 275—1993 has stipulated that the normal load designates the load that is in the range of $0.07 \sim 0.15$ of the ratio of the dynamic equivalent load P and the dynamic load rating C ; the light load designates the load that is smaller than 0.07 of the ratio of the dynamic equivalent load P and the dynamic load rating C ; the heavy load designates the load that is larger than 0.15 of the ratio of the dynamic equivalent load P and the dynamic load rating C .

For the shaft I: on the floating end, considering the cylindrical roller bearing can be floating in the housing bore, so the selecting fit for the inner ring with shaft is K6, the selecting fit for the outer ring with housing bore is H7; on the locating end, the pair of tapered roller bearing selecting fit is as same as the floating end. For the shafts II and III, the selecting fit for every bearing inner ring with shaft is m6 and the selecting fit for every

bearing outer ring with the housing bore is H7.

Table 3 – 95 lists every bearing inner ring and outer ring dimension tolerances and the shaft, housing bore dimension tolerances.

Table 3 – 95 Bearing inner ring, outer ring and the fitting shaft, housing bore dimension tolerances

Shaft	Position	Bearing model	Accuracy	Fit (shaft)	Bearing inner bore Diameter	Fit shaft Diameter	Fit (housing)	Bearing outer ring Diameter	Housing bore
shaft I	A	30307	0	K6	$35 \begin{smallmatrix} 0 \\ -0.012 \end{smallmatrix}$	$35 \begin{smallmatrix} +0.018 \\ +0.002 \end{smallmatrix}$	H7	$72 \begin{smallmatrix} 0 \\ -0.016 \end{smallmatrix}$	$72 \begin{smallmatrix} +0.030 \\ 0 \end{smallmatrix}$
	B	NUP307	0	K6	$35 \begin{smallmatrix} 0 \\ -0.012 \end{smallmatrix}$	$35 \begin{smallmatrix} +0.018 \\ +0.002 \end{smallmatrix}$	H7	$72 \begin{smallmatrix} 0 \\ -0.013 \end{smallmatrix}$	$72 \begin{smallmatrix} +0.030 \\ 0 \end{smallmatrix}$
shaft II	C	30211	0	m6	$50 \begin{smallmatrix} 0 \\ -0.015 \end{smallmatrix}$	$50 \begin{smallmatrix} +0.030 \\ +0.011 \end{smallmatrix}$	H7	$90 \begin{smallmatrix} 0 \\ -0.018 \end{smallmatrix}$	$90 \begin{smallmatrix} +0.035 \\ 0 \end{smallmatrix}$
	D	30211	0	m6	$50 \begin{smallmatrix} 0 \\ -0.015 \end{smallmatrix}$	$50 \begin{smallmatrix} +0.030 \\ +0.011 \end{smallmatrix}$	H7	$90 \begin{smallmatrix} 0 \\ -0.018 \end{smallmatrix}$	$90 \begin{smallmatrix} +0.035 \\ 0 \end{smallmatrix}$
shaft III $Q = 6kN$	E	30213	0	m6	$65 \begin{smallmatrix} 0 \\ -0.015 \end{smallmatrix}$	$65 \begin{smallmatrix} +0.030 \\ +0.011 \end{smallmatrix}$	H7	$120 \begin{smallmatrix} 0 \\ -0.018 \end{smallmatrix}$	$120 \begin{smallmatrix} +0.035 \\ 0 \end{smallmatrix}$
	F	30213	0	m6	$65 \begin{smallmatrix} 0 \\ -0.015 \end{smallmatrix}$	$65 \begin{smallmatrix} +0.030 \\ +0.011 \end{smallmatrix}$	H7	$120 \begin{smallmatrix} 0 \\ -0.018 \end{smallmatrix}$	$120 \begin{smallmatrix} +0.035 \\ 0 \end{smallmatrix}$
shaft III $Q = 9kN$	E	30214	0	m6	$70 \begin{smallmatrix} 0 \\ -0.015 \end{smallmatrix}$	$70 \begin{smallmatrix} +0.030 \\ +0.011 \end{smallmatrix}$	H7	$125 \begin{smallmatrix} 0 \\ -0.020 \end{smallmatrix}$	$125 \begin{smallmatrix} +0.040 \\ 0 \end{smallmatrix}$
	F	30214	0	m6	$70 \begin{smallmatrix} 0 \\ -0.015 \end{smallmatrix}$	$70 \begin{smallmatrix} +0.030 \\ +0.011 \end{smallmatrix}$	H7	$125 \begin{smallmatrix} 0 \\ -0.020 \end{smallmatrix}$	$125 \begin{smallmatrix} +0.040 \\ 0 \end{smallmatrix}$

3) Clearance

On the floating end of shaft I, it is recommended to select the cylindrical roller bearing clearance group, if the self-aligning roller bearings are used on the two ends of shaft III, it is also recommended to select the bearing clearance group. Because the fit of the bearing outer ring with the housing bore is a clearance fit, so the fit has no effect on the bearing clearance. The only considering factor is the interference fit of bearing inner ring on the shaft. Because the interference fit of the inner ring on the shaft causes the the inner ring to expand slightly, the bearing diameter clearance tends to decrease. Therefore, the select on of the bearing clearance group should be according to the clearance decrease value caused by the interference value of the inner ring on the shaft. Table 3 – 96 lists the evaluation value of the clearance decrease value.

Table 3 – 96 The evaluation value of the clearance decrease value

Content	Shaft I support B cylindrical roller bearing	shaft III supports E, F self-aligning roller bearing	according to
inner ring fit	K6	m6	
ring tolerance	$\phi 35 \begin{smallmatrix} 0 \\ -0.012 \end{smallmatrix}$	$\phi 65 \begin{smallmatrix} 0 \\ -0.015 \end{smallmatrix}$	Table3 – 25
shaft tolerance	$\phi 35 \begin{smallmatrix} +0.018 \\ +0.002 \end{smallmatrix}$	$\phi 65 \begin{smallmatrix} +0.030 \\ +0.011 \end{smallmatrix}$	Table3 – 25
nominal interference value $\Delta d/\mu\text{m}$	30	45	minnum bore diameter tolerance adds max Shaft toler
Fitting surface pressed dimension $G^*/\mu\text{m}$	2.5	2.5	Table 3 – 51 (fine grinding)
effective interference amount $\Delta dy/\mu\text{m}$	17.5	27.5	$\Delta d_y = \frac{2}{3}\Delta d - G^*$
d/h	0.76	0.825	Figure 3 – 20c) ,f)
expanding value of inner ring raceway rib diameter $\Delta j/\mu\text{m}$	≈ 13.3	≈ 22.7	Table 3 – 52 equation (1)
clearance group	0 grade mini. 25 μm max 50 μm	0 grade mini 40 μm max 65 μm	reference [12]

The bearing model of the above listed self-aligning bearing clearance group is 22213. When the subjected load of shaft III is 9kN, the bearing model is 22214. Because the tolerance band of the $\phi 65\text{mm}$ is the same as the $\phi 70\text{mm}$, there is no need to reevaluate.

For other supports, all are adjustable supports, their clearance values can be determined according to their using requirements. The clearance of a pair of tapered roller bearings mounted on the locating end of the shaft I can just be adjustable for a little amount in order to increase the support rigidity. The bearing clearance of every support should include the bearing interior clearance and the sparing amount for the shaft expanding caused by the heat, usually taking it as 0.5mm.

The bearing clearance is adjusted all by the washer between the end

cap and the bearing end face. The axial position of the shaft I is adjusted by the snap ring in the housing bore and the adjustable ring on the outer ring end face, which is not convenient as the sleeve cup structure.

(8) Lubricating and sealing

Every bearing can use the oil lubrication, but also can use the grease lubrication. When using the oil lubrication, it should be considered that which channel will be used for the oil supplying.

Concerning the sealing, usually, the contact rubber bowl sealing device can be used.

After finishing the structure design, it also should check the strength of shaft key. Generally, the bearing dimensions are not changed.

Chapter 4 Rolling Bearing Selection and Selection Calculation

4.1 Rolling Bearing Selection Procedure

4.1.1 Common Selection Procedure

1. Common Selection procedure

(1) Select bearing type Selection of bearing type mainly is determined according to the load type, the load direction and the load value.



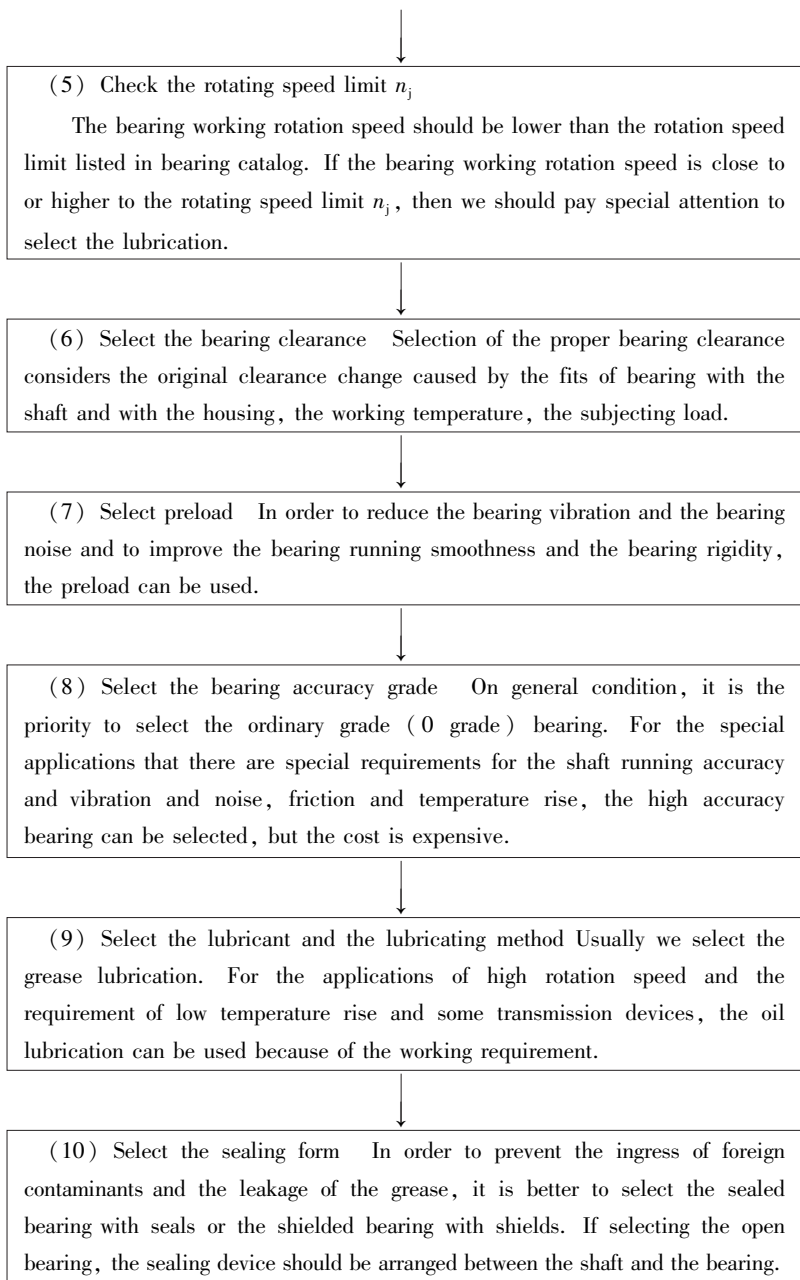
(2) Calculate the value of the dynamic load rating From the actual load on the bearing we calculate the dynamic equivalent load P , and then according to the required life we calculate the required dynamic load rating C .



(3) Preliminarily determine the bearing model we check out the bearing model from the bearing catalog of which the bearing dynamic load rating is close to the required dynamic load rating C , and check out the bearing inner ring diameter, the outer ring diameter, the bearing width and other outside dimensions and the values of the static load rating C_0 and the speed limit n_j .



(4) Check the static load rating C_0 We calculate the static equivalent load P_0 from the actual load on the bearing and check if $P_0 \leq C_0$. If this condition is not met, that is $P_0 > C_0$, then we reselect a bigger dimension bearing. For the applications that the requirement for running accuracy is low and the requirement for smoothly running is not high and there is no shocking and vibration, it is permissible that P_0 can be larger than C_0 for a little bit.



↓

(11) Select the bearing fits with the shaft and with the housing. Selection of the bearing inner bore fit with the shaft and the bearing outer ring fit with the housing considers the load type and the load value.

↓

(12) Check the bearing axial load capacity. Every type of bearing has the different bearing axial load capacity. If the bearing is subjected to axial load, it should be checked that if the axial load is in the permissible value range.

2. To select rolling bearing according to the locating space for the bearing

In many cases, the bearing locating space has been predetermined in machinery design, that means that the bearing inner ring diameter, the bearing outer ring diameter and the bearing width have been determined, the only thing is to select the bearing type, finding the proper bearing with the same dimensions as predetermined, and then check if the selected bearing is the proper one.

3. To check the bearing life for known bearing model

In some cases, the bearing model has been predetermined, but we do not know if the predetermined bearing is the proper one? In this case, we should check the selected bearing model according to the above discussed steps.

4.1.2 Rolling Bearing Type Selection

The following aspects should be taken into account when selecting the bearing type:

1. The load direction, the load value and the load characteristic

All radial bearings can carry the radial load; all thrust bearings can carry the axial load; for the applications of carrying the radial load and the axial load at the same time, the angular contact ball bearing and the tapered roller bearing can be selected. For the applications of just carrying a few axial loads, the deep groove ball bearing can be selected. The angular contact ball bearing and the tapered roller bearing must be used in paired

mounting. Generally, the roller bearing load capacity is larger than the ball bearing with the same dimension and can carry heavier shocking load.

2. The rotation speed

Usually, the bearing rotating speed should be lower than the bearing rotating speed limit n_j . The deep groove ball bearing and the cylindrical roller bearing have higher rotating speed limit and are suitable for using in high speed applications. The thrust bearing rotating speed limit is lower.

3. The requirement limiting locating movement for the support

The bearings that can carry double direction axial loads can be used as the locating supports limiting the double direction axial movements; The bearings that can just carry one direction axial load can be used as the one direction locating supports limiting the one direction axial movements; the floating support can not limit the axial movements, and the non-separable radial bearings which could be moving in the housing bore can be selected and the separable cylindrical bearings in which the inner ring and the outer ring can make moving relative to each other also can be selected.

4. The self-aligning performance

Owing to many reasons, if the coaxality can not be guaranteed between the two housing bores and if the shaft deflection is large, the self-aligning ball bearings or the self-aligning roller bearings can be used because of their good self-aligning performance. The self-aligning capability of the cylindrical roller bearings and the needle roller bearings is just very little.

5. The rigidity requirement

Generally, the rigidity of roller bearings is large and the rigidity of ball bearings is small. The support rigidity of the angular contact ball bearings and the tapered roller bearings can be enhanced through the preload.

6. Others

If the bearing radial space is limited, the needle roller bearings and the needle roller and cage assembly can be used. For the applications that there is a requirement for low vibration and low noise, the low noise deep groove ball bearings can be selected. Besides, the economical condition and the supplying condition should be also taken into account while selecting rolling bearings.

Table 4 – 1 lists varied type bearing performance for reference in selecting rolling bearings

Table 4-1 The varied type bearing performance

Bearing performance	Ball bearing				Roller bearing			
	deep groove ball bearing	angular contact ball bearing	self-aligning ball bearing	thrust ball bearing	cylindrical roller bearing	tapered roller bearing	self-aligning roller bearing	thrust roller bearing
load direction								
axial load capacity	small	larger	very small	larger	small ^①	large	very small	large
radial load capacity (same bore diameter)	1.0	1.0 ~ 1.4	0.6 ~ 0.9	0	1.5 ~ 3.0	1.5 ~ 2.5	2.3 ~ 5.2	0
rotation speed limit	high	high	medium	low	high	medium	low	low
friction coefficient ^②	0.0015 ~ 0.0022	0.0018 ~ 0.0025	0.0010 ~ 0.0018	0.0013 ~ 0.0020	0.0011 ~ 0.0020	0.0018 ~ 0.0028	0.0018 ~ 0.0025	0.0018 ~ 0.0030
rigidity	smaller	smaller	smaller	smaller	larger	larger	larger	larger
manufacturing accuracy	every grade	every grade	0 to 6 grade	every grade	every grade	every grade	0 to 6 grade	0 to 6 grade
vibration and noise	small	smaller	ordinary	ordinary	small	ordinary	ordi.	ordi.
self-aligning	2' ~ 10'	2' ~ 10'	1.5° ~ 3°	—	< 2'	< 2'	1.5° ~ 2.5°	—

Note: Symbol meaning: $\leftarrow$ can carry single direction axial load

$\leftrightarrow$ can carry double direction axial load

$\updownarrow$ can carry radial load

① Cylindrical roller bearing with inner ring rib and outer ring rib, can carry single direction axial load.

② Reference from the machinery hand book No. 29 of vol. 5 of Machinery Press LTD.

4. 1. 3 Selecting Rolling Bearing According to Fatigue Life

In rolling bearings, the contacting surfaces (the raceway surface and the rolling element surface) are subjected to load and also making relative moving, because of the stresses repeatedly acting on these surfaces, at first, the fatigue cracking commences at the material strength weak points below the surface of the raceway material in some depth. This crack gradually propagates to the contacting surface, eventually, forms a flaking dropping off from the raceways or the rolling elements in the load-carrying surface to make the bearing not normally working. In fact, under normally mounted, lubricated, sealed, most of rolling bearings are fatigue failure. So, under ordinary condition, rolling bearings can be selected according to the fatigue life.

1. Rolling bearing fatigue life calculation method

The basic calculation equation of the rolling bearing fatigue life are as follows:

$$\text{for ball bearing:} \quad L = \left(\frac{C}{P} \right)^3 \quad (4-1)$$

$$\text{for roller bearing:} \quad L = \left(\frac{C}{P} \right)^{10/3} \quad (4-2)$$

where L — rating life (million rotations)

C — rating load (kN);

P — dynamic equivalent load (kN).

In the actual calculation, it is inconvenient to use the unit of million rotations, generally, the unit is the working hour. In this case, the equations (4-1) and (4-2) can be rewritten as:

$$\text{for ball bearing:} \quad L_h = \frac{10^6}{60n} \left(\frac{C}{P} \right)^3 \quad (4-3)$$

$$\text{for roller bearing:} \quad L_h = \frac{10^6}{60n} \left(\frac{C}{P} \right)^{10/3} \quad (4-4)$$

where L_h — rating life (h);

n — bearing working rotation speed (r/min).

In the vehicle bearings, usually, the bearing life unit is expressed by

kilometers, In this case, the equations (4-1) and (4-2) can be rewritten as:

$$\text{for ball bearing:} \quad L_k = \pi D \left(\frac{C}{P} \right)^3 \quad (4-5)$$

$$\text{for roller bearing:} \quad L_k = \pi D \left(\frac{C}{P} \right)^{10/3} \quad (4-6)$$

where L_k — rating life (km);

D — wheel diameter (mm).

In order to simplify the bearing life calculation, the rotation speed factor f_n and the life factor f_h will be introduced for getting the calculation result quickly. We also can check the bearing life by checking the tables. The simple calculation tables will be introduced in the following paragraph.

(1) Rotation speed factor f_n and life factor f_h

Introducing the rotation speed factor f_n and the life factor f_h in the equations (4-3) and (4-4):

$$\text{for ball bearing:} \quad f_n = \sqrt[3]{\frac{33 \frac{1}{3}}{n}} \quad (4-7)$$

$$\text{for roller bearing:} \quad f_n = \sqrt[10/3]{\frac{33 \frac{1}{3}}{n}} \quad (4-8)$$

$$\text{for ball bearing:} \quad f_h = \sqrt[3]{\frac{L_h}{500}} \quad (4-9)$$

$$\text{for roller bearing:} \quad f_h = \sqrt[10/3]{\frac{L_h}{500}} \quad (4-10)$$

Therefore, the equations (4-3) and (4-4) can be rewritten as:

$$C = \frac{f_h}{f_n} P \quad (4-11)$$

Table 4-2 lists the life calculation equations for the ball bearing and the roller bearing.

Table 4-3 lists the corresponding values between the rotation speed n and the f_n .

Table 4-4 lists the corresponding values between the bearing life L_h

and the f_h .

Table 4 – 2 Life factor and rotation speed factor calculation equations

Bearing type	Ball bearing	Roller bearing
life/h	$L_h = 500f_h^3$	$L_h = 500f_h^{40/3}$
life factor	$f_h = f_n \frac{C}{P}$	$f_h = f_n \frac{C}{P}$
rotation speed factor	$f_n = \left(\frac{33.3}{n}\right)^{1/3}$	$f_n = \left(\frac{33.3}{n}\right)^{3/10}$

Table 4 – 3 Rotation speed factor f_n

n / (r/min)	f_n		n / (r/min)	f_n		n / (r/min)	f_n	
	ball bearing	roller bearing		ball bearing	roller bearing		ball bearing	roller bearing
10	1. 494	1. 435	25	1. 110	1. 090	40	0. 941	0. 947
11	1. 447	1. 395	26	1. 086	1. 077	41	0. 933	0. 940
12	1. 405	1. 359	27	1. 073	1. 065	42	0. 926	0. 933
13	1. 369	1. 326	28	1. 060	1. 054	43	0. 919	0. 927
14	1. 335	1. 297	29	1. 048	1. 043	44	0. 912	0. 920
15	1. 305	1. 271	30	1. 036	1. 032	45	0. 905	0. 914
16	1. 277	1. 246	31	1. 025	1. 022	46	0. 898	0. 908
17	1. 252	1. 226	32	1. 014	1. 012	47	0. 892	0. 902
18	1. 228	1. 203	33	1. 003	1. 003	48	0. 885	0. 896
19	1. 206	1. 184	34	0. 994	0. 994	49	0. 880	0. 891
20	1. 186	1. 166	35	0. 984	0. 986	50	0. 874	0. 886
21	1. 166	1. 149	36	0. 975	0. 977	52	0. 863	0. 875
22	1. 148	1. 133	37	0. 966	0. 969	54	0. 851	0. 865
23	1. 132	1. 118	38	0. 958	0. 962	56	0. 841	0. 856
24	1. 116	1. 104	39	0. 949	0. 954	58	0. 831	0. 847

(continued)

n / (r/min)	f_n		n / (r/min)	f_n		n / (r/min)	f_n	
	ball bearing	roller bearing		ball bearing	roller bearing		ball bearing	roller bearing
60	0. 822	0. 838	125	0. 644	0. 673	300	0. 481	0. 517
62	0. 813	0. 830	130	0. 635	0. 665	310	0. 476	0. 512
64	0. 805	0. 822	135	0. 627	0. 657	320	0. 471	0. 507
66	0. 797	0. 815	140	0. 620	0. 650	330	0. 466	0. 503
68	0. 788	0. 807	145	0. 613	0. 643	340	0. 461	0. 498
70	0. 781	0. 800	150	0. 606	0. 637	350	0. 457	0. 494
72	0. 774	0. 794	155	0. 599	0. 631	360	0. 453	0. 490
74	0. 767	0. 787	160	0. 593	0. 625	370	0. 448	0. 486
76	0. 760	0. 781	165	0. 586	0. 619	380	0. 444	0. 482
78	0. 753	0. 775	170	0. 581	0. 613	390	0. 441	0. 478
80	0. 747	0. 769	175	0. 575	0. 608	400	0. 437	0. 475
82	0. 741	0. 763	180	0. 570	0. 603	410	0. 433	0. 471
84	0. 735	0. 758	185	0. 565	0. 598	420	0. 430	0. 467
86	0. 729	0. 753	190	0. 560	0. 593	430	0. 426	0. 464
88	0. 724	0. 747	195	0. 555	0. 589	440	0. 423	0. 461
90	0. 718	0. 742	200	0. 550	0. 584	450	0. 420	0. 458
92	0. 713	0. 737	210	0. 541	0. 576	460	0. 417	0. 455
94	0. 708	0. 733	220	0. 533	0. 568	470	0. 414	0. 452
96	0. 703	0. 728	230	0. 525	0. 560	480	0. 411	0. 449
98	0. 698	0. 724	240	0. 518	0. 553	490	0. 408	0. 447
100	0. 693	0. 719	250	0. 511	0. 546	500	0. 406	0. 444
105	0. 682	0. 709	260	0. 504	0. 540	520	0. 400	0. 439
110	0. 672	0. 699	270	0. 498	0. 534	540	0. 395	0. 434
115	0. 662	0. 690	280	0. 492	0. 528	560	0. 390	0. 429
120	0. 652	0. 681	290	0. 487	0. 523	580	0. 386	0. 425

(continued)

n / (r/min)	f_n		n / (r/min)	f_n		n / (r/min)	f_n	
	ball bearing	roller bearing		ball bearing	roller bearing		ball bearing	roller bearing
600	0. 382	0. 420	1250	0. 299	0. 337	3000	0. 223	0. 259
620	0. 378	0. 416	1300	0. 295	0. 333	3100	0. 221	0. 257
640	0. 374	0. 412	1350	0. 291	0. 329	3200	0. 218	0. 254
660	0. 370	0. 408	1400	0. 288	0. 326	3300	0. 216	0. 252
680	0. 366	0. 405	1450	0. 284	0. 322	3400	0. 214	0. 250
700	0. 363	0. 401	1500	0. 281	0. 319	3500	0. 212	0. 248
720	0. 359	0. 398	1550	0. 278	0. 316	3600	0. 210	0. 246
740	0. 356	0. 395	1600	0. 275	0. 313	3700	0. 208	0. 243
760	0. 353	0. 391	1650	0. 272	0. 310	3800	0. 206	0. 242
780	0. 350	0. 388	1700	0. 270	0. 307	3900	0. 205	0. 240
800	0. 347	0. 385	1750	0. 267	0. 305	4000	0. 203	0. 238
820	0. 344	0. 383	1800	0. 265	0. 302	4100	0. 201	0. 236
840	0. 341	0. 380	1850	0. 262	0. 300	4200	0. 199	0. 234
860	0. 339	0. 377	1900	0. 260	0. 297	4300	0. 198	0. 233
880	0. 336	0. 375	1950	0. 258	0. 295	4400	0. 196	0. 231
900	0. 333	0. 372	2000	0. 255	0. 293	4500	0. 195	0. 230
920	0. 331	0. 370	2100	0. 251	0. 289	4600	0. 193	0. 228
940	0. 329	0. 367	2200	0. 247	0. 285	4700	0. 192	0. 227
960	0. 326	0. 365	2300	0. 244	0. 281	4800	0. 191	0. 225
980	0. 324	0. 363	2400	0. 240	0. 277	4900	0. 190	0. 224
1000	0. 322	0. 361	2500	0. 237	0. 274	5000	0. 188	0. 222
1050	0. 317	0. 355	2600	0. 234	0. 271	5200	0. 186	0. 220
1100	0. 312	0. 350	2700	0. 231	0. 268	5400	0. 183	0. 217
1150	0. 307	0. 346	2800	0. 228	0. 265	5600	0. 181	0. 215
1200	0. 303	0. 341	2900	0. 226	0. 262	5800	0. 179	0. 213

(continued)

n / (r/min)	f_n		n / (r/min)	f_n		n / (r/min)	f_n	
	ball bearing	roller bearing		ball bearing	roller bearing		ball bearing	roller bearing
6000	0. 177	0. 211	9600	0. 152	0. 183	17500	0. 124	0. 153
6200	0. 175	0. 209	9800	0. 150	0. 182	18000	0. 123	0. 152
6400	0. 173	0. 207				18500	0. 122	0. 150
6600	0. 172	0. 205	10000	0. 149	0. 181	19000	0. 121	0. 149
6800	0. 170	0. 203	10500	0. 147	0. 178	19500	0. 120	0. 148
			11000	0. 145	0. 176			
7000	0. 168	0. 201	11500	0. 143	0. 173	20000	0. 119	0. 147
7200	0. 167	0. 199	12000	0. 141	0. 171	21000	0. 117	0. 145
7400	0. 165	0. 198				22000	0. 115	0. 143
7600	0. 164	0. 196	12500	0. 139	0. 169	23000	0. 113	0. 141
7800	0. 162	0. 195	13000	0. 137	0. 167	24000	0. 112	0. 139
			13500	0. 135	0. 165			
8000	0. 161	0. 193	14000	0. 134	0. 163	25000	0. 110	0. 137
8200	0. 160	0. 192	14500	0. 132	0. 162	26000	0. 109	0. 136
8400	0. 158	0. 190				27000	0. 107	0. 134
8600	0. 157	0. 189	15000	0. 131	0. 160	28000	0. 106	0. 133
8800	0. 156	0. 188	15500	0. 129	0. 158	29000	0. 105	0. 131
			16000	0. 128	0. 157			
9000	0. 155	0. 187	16500	0. 126	0. 156	30000	0. 104	0. 130
9200	0. 154	0. 185	17000	0. 125	0. 154			
9400	0. 153	0. 184						

Table 4 -4 Life factor f_h

L_h /h	f_h		L_h /h	f_h		L_h /h	f_h	
	ball bearing	roller bearing		ball bearing	roller bearing		ball bearing	roller bearing
100	0. 585	0. 617	110	0. 604	0. 635	120	0. 622	0. 652
105	0. 595	0. 626	115	0. 613	0. 643	125	0. 631	0. 660

(continued)

L_h /h	f_h		L_h /h	f_h		L_h /h	f_h	
	ball bearing	roller bearing		ball bearing	roller bearing		ball bearing	roller bearing
130	0. 639	0. 668	310	0. 852	0. 866	620	1. 075	1. 065
135	0. 647	0. 675	320	0. 861	0. 875	640	1. 085	1. 075
140	0. 654	0. 683	330	0. 870	0. 883	660	1. 100	1. 085
145	0. 662	0. 690	340	0. 879	0. 891	680	1. 110	1. 095
150	0. 670	0. 697	350	0. 888	0. 898	700	1. 120	1. 105
155	0. 677	0. 704	360	0. 896	0. 906	720	1. 130	1. 115
160	0. 684	0. 710	370	0. 905	0. 914	740	1. 140	1. 125
165	0. 691	0. 717	380	0. 913	0. 921	760	1. 150	1. 135
170	0. 698	0. 723	390	0. 921	0. 928	780	1. 160	1. 145
175	0. 705	0. 730	400	0. 928	0. 935	800	1. 170	1. 150
180	0. 712	0. 736	410	0. 936	0. 942	820	1. 180	1. 160
185	0. 718	0. 742	420	0. 944	0. 949	840	1. 190	1. 170
190	0. 724	0. 748	430	0. 951	0. 956	860	1. 200	1. 180
195	0. 731	0. 754	440	0. 959	0. 962	880	1. 205	1. 185
200	0. 737	0. 760	450	0. 966	0. 969	900	1. 215	1. 190
210	0. 749	0. 771	460	0. 973	0. 975	920	1. 225	1. 200
220	0. 761	0. 782	470	0. 980	0. 982	940	1. 235	1. 210
230	0. 772	0. 792	480	0. 987	0. 988	960	1. 245	1. 215
240	0. 783	0. 802	490	0. 994	0. 994	980	1. 250	1. 225
250	0. 794	0. 812	500	1. 000	1. 000	1000	1. 260	1. 230
260	0. 804	0. 822	520	1. 015	1. 010	1050	1. 280	1. 250
270	0. 814	0. 831	540	1. 025	1. 025	1100	1. 300	1. 270
280	0. 824	0. 840	560	1. 040	1. 035	1150	1. 320	1. 285
290	0. 834	0. 849	580	1. 050	1. 045	1200	1. 340	1. 300
300	0. 843	0. 858	600	1. 065	1. 055	1250	1. 360	1. 315

(continued)

L_h /h	f_h		L_h /h	f_h		L_h /h	f_h	
	ball bearing	roller bearing		ball bearing	roller bearing		ball bearing	roller bearing
1300	1. 375	1. 330	3100	1. 835	1. 730	6200	2. 32	2. 13
1350	1. 395	1. 345	3200	1. 855	1. 745	6400	2. 34	2. 15
1400	1. 410	1. 360	3300	1. 875	1. 760	6600	2. 37	2. 17
1450	1. 425	1. 375	3400	1. 895	1. 775	6800	2. 39	2. 19
1500	1. 445	1. 390	3500	1. 910	1. 795	7000	2. 41	2. 21
1550	1. 460	1. 405	3600	1. 930	1. 810	7200	2. 43	2. 23
1600	1. 475	1. 420	3700	1. 950	1. 825	7400	2. 46	2. 24
1650	1. 490	1. 430	3800	1. 965	1. 840	7600	2. 48	2. 26
1700	1. 505	1. 445	3900	1. 985	1. 850	7800	2. 50	2. 28
1750	1. 520	1. 455	4000	2. 00	1. 865	8000	2. 52	2. 30
1800	1. 535	1. 470	4100	2. 02	1. 880	8200	2. 54	2. 31
1850	1. 545	1. 480	4200	2. 03	1. 895	8400	2. 56	2. 33
1900	1. 560	1. 490	4300	2. 05	1. 905	8600	2. 58	2. 35
1950	1. 575	1. 505	4400	2. 07	1. 920	8800	2. 60	2. 36
2000	1. 590	1. 515	4500	2. 08	1. 935	9000	2. 62	2. 38
2100	1. 615	1. 540	4600	2. 10	1. 945	9200	2. 64	2. 40
2200	1. 640	1. 560	4700	2. 11	1. 960	9400	2. 66	2. 41
2300	1. 665	1. 580	4800	2. 13	1. 970	9600	2. 68	2. 43
2400	1. 690	1. 600	4900	2. 14	1. 985	9800	2. 70	2. 44
2500	1. 710	1. 620	5000	2. 15	2. 00	10000	2. 71	2. 46
2600	1. 730	1. 640	5200	2. 18	2. 02	10500	2. 76	2. 49
2700	1. 755	1. 660	5400	2. 21	2. 04	11000	2. 80	2. 53
2800	1. 775	1. 675	5600	2. 24	2. 06	11500	2. 85	2. 56
2900	1. 795	1. 695	5800	2. 27	2. 09	12000	2. 89	2. 59
3000	1. 815	1. 710	6000	2. 29	2. 11	12500	2. 93	2. 63

(continued)

L_h /h	f_h		L_h /h	f_h		L_h /h	f_h	
	ball bearing	roller bearing		ball bearing	roller bearing		ball bearing	roller bearing
13000	2. 96	2. 66	25000	3. 68	3. 23	43000	4. 42	3. 80
13500	3. 00	2. 69				44000	4. 45	3. 83
14000	3. 04	2. 72	26000	3. 73	3. 27	45000	4. 48	3. 86
14500	3. 07	2. 75	27000	3. 78	3. 31			
15000	3. 11	2. 77	28000	3. 82	3. 35	46000	4. 51	3. 88
			29000	3. 87	3. 38	47000	4. 55	3. 91
15500	3. 14	2. 80	30000	3. 91	3. 42	48000	4. 58	3. 93
16000	3. 18	2. 83				49000	4. 61	3. 96
16500	3. 21	2. 85	31000	3. 96	3. 45	50000	4. 64	3. 98
17000	3. 24	2. 88	32000	4. 00	3. 48			
17500	3. 27	2. 91	33000	4. 04	3. 51	55000	4. 80	4. 10
			34000	4. 08	3. 55	60000	4. 94	4. 20
18000	3. 30	2. 93	35000	4. 12	3. 58	65000	5. 07	4. 30
18500	3. 33	2. 95				70000	5. 19	4. 40
19000	3. 36	2. 98	36000	4. 16	3. 61	75000	5. 30	4. 50
19500	3. 39	3. 00	37000	4. 20	3. 64			
20000	3. 42	3. 02	38000	4. 24	3. 67	80000	5. 43	4. 58
			39000	4. 27	3. 70	85000	5. 55	4. 66
21000	3. 48	3. 07	40000	4. 31	3. 72	90000	5. 65	4. 75
22000	3. 53	3. 11				100000	5. 85	4. 90
23000	3. 58	3. 15	41000	4. 35	3. 75			
24000	3. 63	3. 19	42000	4. 38	3. 78			

Note: L_h is the bearing rating life.

(2) From the known Bearing life L_h and bearing rotation speed, we can check out directly the value of C/P .

From the equation (4-11) knowing that the value of the dynamic equivalent load can be calculated from the load subjected on the bearing. Therefore, if knowing the bearing rotation speedn and the required beared life L_h , the required C/P value for ball bearings can be easily checked out from Table 4 - 5, and from Table 4 - 6 we can checke out the required C/P value for roller bearings. Because the P value has been calculated, the C

value can be calculated, and then the required bearing model can be checked out from the bearing catalog.

Table 4-5 Corresponding C/P value with L_h and n (ball bearing)

$n/$ (r/min)	10	16	25	40	63	100	125	160	200
L_h/h	ball bearing C/P								
100									
500				1.06	1.24	1.45	1.56	1.68	1.82
1000			1.15	1.34	1.56	1.82	1.96	2.12	2.29
1250		1.06	1.24	1.45	1.68	1.96	2.12	2.29	2.47
1600		1.15	1.34	1.56	1.82	2.12	2.29	2.47	2.67
2000	1.06	1.24	1.45	1.68	1.96	2.29	2.47	2.67	2.88
2500	1.15	1.34	1.56	1.82	2.12	2.47	2.67	2.88	3.11
3200	1.24	1.45	1.68	1.96	2.29	2.67	2.88	3.11	3.36
4000	1.34	1.56	1.82	2.12	2.47	2.88	3.11	3.36	3.63
5000	1.45	1.68	1.96	2.29	2.67	3.11	3.36	3.63	3.91
6300	1.56	1.82	2.12	2.47	2.88	3.36	3.63	3.91	4.23
8000	1.68	1.96	2.29	2.67	3.11	3.63	3.91	4.23	4.56
10000	1.82	2.12	2.47	2.88	3.36	3.91	4.23	4.56	4.93
12500	1.96	2.29	2.67	3.11	3.63	4.23	4.56	4.93	5.32
16000	2.12	2.47	2.88	3.36	3.91	4.56	4.93	5.32	5.75
20000	2.29	2.67	3.11	3.63	4.23	4.93	5.32	5.75	6.20
25000	2.47	2.88	3.36	3.91	4.56	5.32	5.75	6.20	6.70
32000	2.67	3.11	3.63	4.23	4.93	5.75	6.20	6.70	7.23
40000	2.88	3.36	3.91	4.56	5.32	6.20	6.70	7.23	7.81
50000	3.11	3.63	4.23	4.93	5.75	6.70	7.23	7.81	8.43
63000	3.36	3.91	4.56	5.32	6.20	7.23	7.81	8.43	9.11
80000	3.63	4.23	4.93	5.75	6.70	7.81	8.43	9.11	9.83
100000	3.91	4.56	5.32	6.20	7.23	8.43	9.11	9.83	10.6
200000	4.93	5.75	6.70	7.81	9.11	10.6	11.5	12.4	13.4

(continued)

$n/ \text{ (r/min)}$	250	320	400	500	630	800	1000	1250	1600	2000
L_h/h	ball bearing C/P									
100	1. 15	1. 24	1. 34	1. 45	1. 56	1. 68	1. 82	1. 96	2. 12	2. 29
500	1. 96	2. 12	2. 29	2. 47	2. 67	2. 88	3. 11	3. 36	3. 63	3. 91
1000	2. 47	2. 67	2. 88	3. 11	3. 36	3. 63	3. 91	4. 23	4. 56	4. 93
1250	2. 67	2. 88	3. 11	3. 36	3. 63	3. 91	4. 23	4. 56	4. 93	5. 32
1600	2. 88	3. 11	3. 36	3. 63	3. 91	4. 23	4. 56	4. 93	5. 32	5. 75
2000	3. 11	3. 36	3. 63	3. 91	4. 23	4. 56	4. 93	5. 32	5. 75	6. 20
2500	3. 36	3. 63	3. 91	4. 23	4. 56	4. 93	5. 32	5. 75	6. 20	6. 70
3200	3. 63	3. 91	4. 23	4. 56	4. 93	5. 32	5. 75	6. 20	6. 70	7. 23
4000	3. 91	4. 23	4. 56	4. 93	5. 32	5. 75	6. 20	6. 70	7. 23	7. 81
5000	4. 23	4. 56	4. 93	5. 32	5. 75	6. 20	6. 70	7. 23	7. 81	8. 43
6300	4. 56	4. 93	5. 32	5. 75	6. 20	6. 70	7. 23	7. 81	8. 43	9. 11
8000	4. 93	5. 32	5. 75	6. 20	6. 70	7. 23	7. 81	8. 43	9. 11	9. 83
10000	5. 32	5. 75	6. 20	6. 70	7. 23	7. 81	8. 43	9. 11	9. 83	10. 6
12500	5. 75	6. 20	6. 70	7. 23	7. 81	8. 43	9. 11	9. 83	10. 6	11. 5
16000	6. 20	6. 70	7. 23	7. 81	8. 43	9. 11	9. 83	10. 6	11. 5	12. 4
20000	6. 70	7. 23	7. 81	8. 43	9. 11	9. 83	10. 6	11. 5	12. 4	13. 4
25000	7. 23	7. 81	8. 43	9. 11	9. 83	10. 6	11. 5	12. 4	13. 4	14. 5
32000	7. 81	8. 43	9. 11	9. 83	10. 6	11. 5	12. 4	13. 4	14. 5	15. 6
40000	8. 43	9. 11	9. 83	10. 6	11. 5	12. 4	13. 4	14. 5	15. 6	16. 8
50000	9. 11	9. 83	10. 6	11. 5	12. 4	13. 4	14. 5	15. 6	16. 8	18. 2
63000	9. 83	10. 6	11. 5	12. 4	13. 4	14. 5	15. 6	16. 8	18. 2	19. 6
80000	10. 6	11. 5	12. 4	13. 4	14. 5	15. 6	16. 8	18. 2	19. 6	21. 2
100000	11. 5	12. 4	13. 4	14. 5	15. 6	16. 8	18. 2	19. 6	21. 2	22. 9
200000	14. 5	15. 6	16. 8	18. 2	19. 6	21. 2	22. 9	24. 7	26. 7	28. 8

(continued)

$n/(\text{r/min})$	2500	3200	4000	5000	6300	8000	10000	12500	16000
L_h/h	ball bearing C/P								
100	2.47	2.67	2.88	3.11	3.36	3.63	3.91	4.23	4.56
500	4.23	4.56	4.93	5.32	5.75	6.20	6.70	7.23	7.81
1000	5.32	5.75	6.20	6.70	7.23	7.81	8.43	9.11	9.83
1250	5.75	6.20	6.70	7.23	7.81	8.43	9.11	9.83	10.6
1600	6.20	6.70	7.23	7.81	8.43	9.11	9.83	10.6	11.5
2000	6.70	7.23	7.81	8.43	9.11	9.83	10.6	11.5	12.4
2500	7.23	7.81	8.43	9.11	9.83	10.6	11.5	12.4	13.4
3200	7.81	8.43	9.11	9.83	10.6	11.5	12.4	13.4	14.5
4000	8.43	9.11	9.83	10.6	11.5	12.4	13.4	14.5	15.6
5000	9.11	9.83	10.6	11.5	12.4	13.4	14.5	15.6	16.8
6300	9.83	10.6	11.5	12.4	13.4	14.5	15.6	16.8	18.2
8000	10.6	11.5	12.4	13.4	14.5	15.6	16.8	18.2	19.6
10000	11.5	12.4	13.4	14.5	15.6	16.8	18.2	19.6	21.2
12500	12.4	13.4	14.5	15.6	16.8	18.2	19.6	21.2	22.9
16000	13.4	14.5	15.6	16.8	18.2	19.6	21.2	22.9	24.7
20000	14.5	15.6	16.8	18.2	19.6	21.2	22.9	24.7	26.7
25000	15.6	16.8	18.2	19.6	21.2	22.9	24.7	26.7	28.8
32000	16.8	18.2	19.6	21.2	22.9	24.7	26.7	28.8	31.1
40000	18.2	19.6	21.2	22.9	24.7	26.7	28.8	31.1	
50000	19.6	21.2	22.9	24.7	26.7	28.2	31.1		
53000	21.2	22.9	24.7	26.7	28.8	31.1			
80000	22.9	24.7	26.7	28.8	31.1				
100000	24.7	26.7	28.8	31.1					
200000	31.1								

Table 4-6 Corresponding C/P value with L_h and n (roller bearing)

$n/$ (r/min)	10	16	25	40	63	100	125	160	200
L_h/h	roller bearing C/P								
100									1.05
500				1.05	1.21	1.39	1.49	1.60	1.71
1000			1.13	1.30	1.49	1.71	1.83	1.97	2.11
1250		1.05	1.21	1.39	1.60	1.83	1.97	2.11	2.26
1600		1.13	1.30	1.49	1.71	1.97	2.11	2.26	2.42
2000	1.05	1.21	1.39	1.60	1.83	2.11	2.26	2.42	2.59
2500	1.13	1.30	1.49	1.71	1.97	2.26	2.42	2.59	2.78
3200	1.21	1.39	1.60	1.83	2.11	2.42	2.59	2.78	2.97
4000	1.30	1.49	1.71	1.97	2.26	2.59	2.78	2.97	3.19
5000	1.39	1.60	1.83	2.11	2.42	2.78	2.97	3.19	3.42
6300	1.49	1.71	1.97	2.26	2.59	2.97	3.19	3.42	3.66
8000	1.6	1.83	2.11	2.42	2.78	3.19	3.42	3.66	3.92
10000	1.71	1.97	2.26	2.59	2.97	3.42	3.66	3.92	4.20
12500	1.83	2.11	2.42	2.78	3.19	3.66	3.92	4.20	4.50
16000	1.97	2.26	2.59	2.97	3.42	3.92	4.20	4.50	4.82
20000	2.11	2.42	2.78	3.19	3.66	4.20	4.50	4.82	5.17
25000	2.26	2.59	2.97	3.42	3.92	4.50	4.82	5.17	5.54
32000	2.42	2.78	3.19	3.66	4.20	4.82	5.17	5.54	5.94
40000	2.59	2.97	3.42	3.92	4.50	5.17	5.54	5.94	6.36
50000	2.78	3.19	3.66	4.20	4.82	5.54	5.94	6.36	6.81
63000	2.97	3.42	3.92	4.50	5.17	5.94	6.36	6.81	7.30
80000	3.19	3.66	4.20	4.82	5.54	6.36	6.81	7.30	7.82
100000	3.42	3.92	4.50	5.17	5.94	6.81	7.30	7.82	8.38
200000	4.20	4.82	5.54	6.36	7.30	8.38	8.98	9.62	10.3

(continued)

$n/$ (r/min)	250	320	400	500	630	800	1000	1250	1600
L_h/h	roller bearing C/P								
100	1.13	1.21	1.30	1.39	1.49	1.6	1.71	1.83	1.97
500	1.83	1.97	2.11	2.26	2.42	2.59	2.78	2.97	3.19
1000	2.26	2.42	2.59	2.78	2.97	3.19	3.42	3.66	3.92
1250	2.42	2.59	2.78	2.97	3.19	3.42	3.66	3.92	4.20
1600	2.59	2.78	2.97	3.19	3.42	3.66	3.92	4.20	4.50
2000	2.78	2.97	3.19	3.42	3.66	3.92	4.20	4.50	4.82
2500	2.97	3.19	3.42	3.66	3.92	4.20	4.50	4.82	5.17
3200	3.19	3.42	3.66	3.92	4.20	4.50	4.82	5.17	5.54
4000	3.42	3.66	3.92	4.20	4.50	4.82	5.17	5.54	5.94
5000	3.66	3.92	4.20	4.50	4.82	5.17	5.54	5.94	6.36
6300	3.92	4.20	4.50	4.82	5.17	5.54	5.94	6.36	6.81
8000	4.20	4.50	4.82	5.17	5.54	5.94	6.36	6.81	7.30
10000	4.50	4.82	5.17	5.54	5.94	6.36	6.81	7.30	7.82
12500	4.82	5.17	5.54	5.94	6.36	6.81	7.30	7.82	8.38
16000	5.17	5.54	5.94	6.36	6.81	7.30	7.82	8.38	8.98
20000	5.54	5.94	6.36	6.81	7.30	7.82	8.38	8.98	9.62
25000	5.94	6.36	6.81	7.30	7.82	8.38	8.98	9.62	10.3
32000	6.36	6.81	7.30	7.82	8.38	8.98	9.62	10.3	11.0
40000	6.81	7.30	7.82	8.38	8.98	9.62	10.3	11.0	11.8
50000	7.30	7.82	8.38	8.98	9.62	10.3	11.0	11.8	12.7
63000	7.82	8.38	8.98	9.62	10.3	11.0	11.8	12.7	13.6
80000	8.38	8.98	9.62	10.3	11.0	11.8	12.7	13.6	14.6
100000	8.98	9.62	10.3	11.0	11.8	12.7	13.6	14.6	15.6
200000	11.0	11.8	12.7	13.6	14.6	15.6	16.7	17.9	19.2

Table 4-7 Corresponding C/P value with L

L /million r	C/P		L /million r	C/P		L /million r	C/P	
	ball bearing	roller bearing		ball bearing	roller bearing		ball bearing	roller bearing
0.5	0.793	0.812	260	6.38	5.30	2200	13.0	10.1
0.75	0.909	0.917	280	6.54	5.42	2400	13.4	10.3
1.0	1.0	1.0	300	6.69	5.54	2600	13.8	10.6
1.5	1.14	1.13	320	6.84	5.64	2800	14.1	10.8
2.0	1.26	1.24	340	6.98	5.75	3000	14.4	11.0
3.0	1.44	1.39	360	7.11	5.85	3200	14.7	11.3
4.0	1.59	1.52	380	7.24	5.94	3400	15.0	11.5
5.0	1.71	1.62	400	7.37	6.03	3600	15.3	11.7
6.0	1.82	1.71	420	7.49	6.12	3800	15.6	11.9
8.0	2.0	1.87	440	7.61	6.21	4000	15.9	12.0
10	2.15	2.0	460	7.72	6.29	4500	16.5	12.5
12	2.29	2.11	480	7.83	6.37	5000	17.1	12.9
14	2.41	2.21	500	7.94	6.45	5500	17.7	13.2
16	2.52	2.30	550	8.19	6.64	6000	18.2	13.6
18	2.62	2.38	600	8.43	6.81	6500	18.7	13.9
20	2.71	2.46	650	8.66	6.98	7000	19.1	14.2
25	2.92	2.63	700	8.88	7.14	7500	19.6	14.5
30	3.11	2.77	750	9.09	7.29	8000	20.0	14.8
35	3.27	2.91	800	9.28	7.43	8500	20.4	15.1
40	3.42	3.02	850	9.47	7.56	9000	20.8	15.4
45	3.56	3.13	900	9.65	7.70	9500	21.2	15.6
50	3.68	3.23	950	9.83	7.82	10000	21.5	15.8
60	3.91	3.42	1000	10.0	7.94	12000	22.9	16.7
70	4.12	3.58	1100	10.3	8.17	14000	24.1	17.5
80	4.31	3.72	1200	10.6	8.39	16000	25.2	18.2
90	4.48	3.86	1300	10.9	8.59	18000	26.2	18.9
100	4.64	3.98	1400	11.2	8.79	20000	27.1	19.5
120	4.93	4.20	1500	11.4	8.97	25000	29.2	20.9
140	5.19	4.40	1600	11.7	9.15	30000	31.1	22.0
160	5.43	4.58	1700	11.9	9.31			
180	5.65	4.75	1800	12.2	9.48			
200	5.85	4.90	1900	12.4	9.63			
220	6.04	5.04	2000	12.6	9.78			
240	6.21	5.18						

(3) From the known bearing life (million rotations), we can check out the C/P value. If knowing the bearing life L (unit as million rotations), then the required C/P value can be quickly checked out from Table 4-7 or Figure 4-1, and then the required bearing model can be checked out from the bearing catalog.

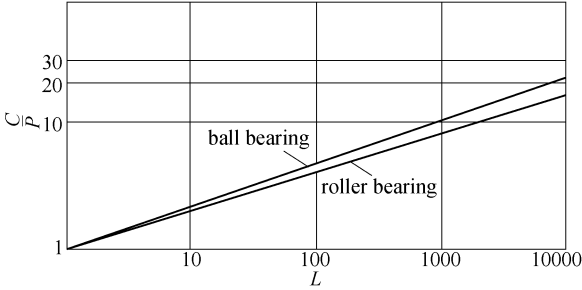


Figure 4-1 Relationship of $\frac{C}{P}$ with L

2. Rolling bearing application life

When selecting rolling bearing, the first thing is to reasonably put forward the required bearing life. If the required bearing life is too long, then the selected bearing dimension would be large, the machinery would be cumbersome and the cost must not be economical. If the required bearing life is too short, then during the serving duration, the selected bearing have to be dismantled and changed often. Usually, the required bearing life should be selected according to the major machinery maintainance period. Table 4-8 lists the recommended bearing application life for every application condition.

Table 4-8 Recommended bearing application life

Application condition	Required bearing life/h
not often used instrument and equipment	300 ~ 3000
short time used or interrupted used machinery, there is no serious result for the interruption, such as manually operated machine, agricultural machinery, mounting crane, automatically supplying equipment	3000 ~ 8000

(continued)

Application condition	Required bearing life/h
interrupted used machinery, there would be serious result for the interruption, such as auxiliary equipment of the electricity plant, continuous working transmission equipment, belt transmission machine, workshop crane	8000 ~ 12000
machine working 8h everyday, but always working under not full load, such as motor, ordinary gear device, crash machine, crane and ordinary machine	10000 ~ 25000
every day work 8h, but full load working, such as machine tool, wood machining machine, engineering machine, printing machine, separating machine, centrifugal machine	20000 ~ 30000
24h continuous working machine, compressor, pump, motor, rolling mill gear device, textile machine	40000 ~ 50000
24h continuous working machine, there would be serious result for the interruption, such as fiber machine, paper machine, electricity station main equipment, supplying and draining equipment, mine pump, mine ventilation machine	about 100000

The recommended bearing lives for varied machine are listed in Table 4 – 8. Table 4 – 9 and Table 4 – 10 recommerd the f_h valve for varied machines when selecting bearings and under varied working conditions, respectively. The recommendel bearing lives in Tables 4 – 8, 4 – 9 and 4 – 10 are based on prior experience accumulated for many years by the foreign bearing companies, but we should refer to many experiences in china while selecting bearing application life.

Table 4 – 9 Recommended f_h values

Mounting position	f_h	Mounting position	f_h
Ship		ship shaft supporting bearing	4. 0 ~ 6. 0
ship thrust bearing	2. 9 ~ 3. 6	ship moving equipment	1. 5 ~ 2. 5
small ship		large ship transmission device	2. 6 ~ 3. 5
transmission device	1. 5 ~ 2. 5		

(continued)

Mounting position	f_h	Mounting position	f_h
automobile		Agricultural machine	
car	1.2 ~ 1.5	tractor	1.5 ~ 2.0
motorcycle	0.7 ~ 1.4	farming machine	1.3 ~ 1.8
passenger car	0.9 ~ 1.4	season fa ming machine	1.0 ~ 1.5
light truck	1.5 ~ 2.0	Construction machine	
heavy truck	1.6 ~ 2.4	caterpillar mounted excavator	1.7 ~ 2.2
bus	1.5 ~ 1.24	excavator walking machinery	1.0 ~ 1.3
internal-combustion engine	1.0 ~ 1.5	excavator rotating machinery	1.4 ~ 1.8
Railway vehicle		vibration motor	1.0 ~ 2.0
transport mine vehicle box	2.5 ~ 3.5	Motor	
track trolley shaft box	3.5 ~ 4.0	home motor	1.5 ~ 2.0
passenger vehicle box	3.0 ~ 3.5	standard motor	3.5 ~ 4.5
transport mine vehicle box	3.0 ~ 3.5	large motor	4.0 ~ 5.0
vehicle shaft box bearing	3.5 ~ 4.0	tractor motor	3.0 ~ 3.5
locomotive shaft box (outside)	3.5 ~ 4.0	Ordinary transmission	
locomotive shaft box (inside)	4.5 ~ 5.0	universal transmission	2.0 ~ 3.0
railway vehicle transmission	3.0 ~ 4.5	transmission gear motor	2.0 ~ 3.0
Milling, metallurgy device	2.9 ~ 3.6	large fixed transmission	3.0 ~ 4.5
milling	2.0 ~ 2.5	centrifugal cast machine	3.4 ~ 4.0
roller track	2.5 ~ 3.5		
milling machine spindle	3.0 ~ 4.0		

(continued)

Mounting position	f_h	Mounting position	f_h
Machine tool		Excavate transportation	
drilling machine spindle	2. 7 ~ 4. 0	outdoor mine transporation equipment	4. 5 ~ 5. 5
grinding machine grinding wheel	2. 3 ~ 3. 4	ordinary transportation band roller	2. 7 ~ 3. 4
machine tool transmission	2. 8 ~ 4. 0	band roller barrel	3. 8 ~ 4. 5
pressure machine fly wheel	3. 4 ~ 4. 0	wheel excavator walking equipment	2. 0 ~ 3. 5
pressure machine eccentric shaft	2. 5 ~ 3. 0	wheel excavator wheel	4. 5 ~ 6. 0
electrical instrument	1. 8 ~ 2. 7	elevator rope wheel	4. 0 ~ 4. 5
wind driven instrument	1. 8 ~ 2. 7	Pump, air blower, air compressor	
Wood machining machine		electric fan , air blower	3. 0 ~ 5. 5
frame sawing machine main bearing	3. 4 ~ 4. 0	centrifugal pump	3. 0 ~ 4. 5
frame sawing machine crank bearing	2. 1 ~ 2. 8	hydro axial cylinder pump	1. 0 ~ 2. 5

Table 4 – 10 Recommended f_h values for working condition

Working condition	Affecting machine working extent by changing bearing	
	no large effect	large effect
short time work	1. 0 ~ 2. 5	2. 0 ~ 3. 5
8h work	2. 0 ~ 4. 0	3. 0 ~ 4. 5
continuous work	3. 5 ~ 5. 0	4. 0 ~ 5. 5

3. Dynamic equivalent load P calculation

If the radial load and the axial load are subjected on the bearing at the same time, in this case, the actual loads must be transformed to the dynamic equivalent load P and only then, the bearing life calculation can be carried on. The dynamic equivalent load is an assumed load, under the dynamic equivalent load, the bearing life is the same as the actual loads acting. Especially for the radial ball bearings and the angular contact ball bearings, the bearing contact angle changing caused by the actual axial load should also be considered while calculating the dynamic equivalent load P . The details concerning the dynamic equivalent load P have been discussed in section 2.6.4.

(1) Rolling bearing equal life curve and dynamic equivalent load P calculation equation According to the Lundberg- Palmgren rolling bearing life theory, the varied type bearing equal life curves can be drawn as shown in Figure 4 - 2. The abscissa is expressed by $\frac{F_a \cot \alpha}{P}$ and the ordinate is expressed by $\frac{F_r}{P}$. The varied combination of the radial load and the axial load on the equal life curve has the same bearing life. In order to simplify the dynamic equivalent load P calculation, the equal life curve is expressed by the straight lines AB , BC instead of the curve AC . The point A coordinates are as: $\frac{F_r}{P} = 1$, $\frac{F_a \cot \alpha}{P} = 1.216$ (ball bearing) or 1.266 (roller bearing). Therefore, we can get:

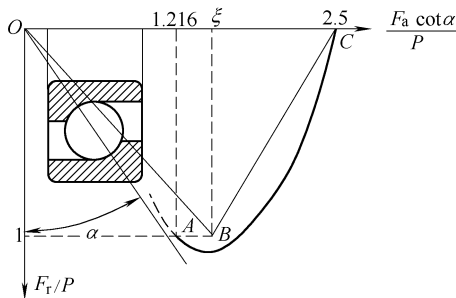


Figure 4 - 2 Bearing equal life curve

for ball bearing:

$$\frac{F_a}{F_r} = 1.216 \tan \alpha$$

for roller bearing:

$$\frac{F_a}{F_r} = 1.266 \tan \alpha$$

This condition is just conformed with the condition of the half raceway loading in rolling bearing (see Section 2.3).

The point B coordinates are as:

$$\frac{F_r}{P} = 1$$

$$\frac{F_a \cot \alpha}{P} = \xi$$

ξ is the factor related to the bearing structure and the bearing type. For ball bearing: $\xi = 1.1$; for angular contact ball bearing: $\xi = 1.25$; for tapered roller bearing: $\xi = 1.5$. If the ratio of F_a and F_r is a little larger than the value of the half raceway loading in rolling bearing, there is no effect for bearing life. If the ratio of F_a and F_r is smaller than the value of the half raceway loading in rolling bearing, the bearing can not normally work, therefore, this part of the curve is expressed by dotted line.

It can be got from Figure 4-2 that the dynamic equivalent load P simplifying calculation equation can be expressed by the straight line equations by using the judgment factor e , the factor e expresses the condition of intersection point of straight lines AB and BC :

$$\left. \begin{array}{ll} \text{While } \frac{F_a}{F_r} \leq e & P = F_r \\ \text{While } \frac{F_a}{F_r} > e & P = XF_r + YF_a \end{array} \right\} \quad (4-12)$$

where X — radial factor;

Y — axial factor;

e — judgment factor, related to bearing contact angle;

α — bearing actual contact angle.

The ball bearing contact angle α value changes with the axial load, but

the contact angle α value can be found indirectly from the $\frac{F_a}{C_0}$ value. The details can be seen from section 2.6.

The concrete value of the radial factor X , the axial factor Y and the judgment factor e for every bearing model can be found from the rolling bearing catalog. Knowing these parameters, we can calculate the equivalent dynamic load P .

The dynamic equivalent load P calculation for the deep groove ball bearing and the angular contact ball bearing must consider the contact angle change under axial load, the calculation steps are as follows:

1) Calculate the $\frac{F_a}{C_0}$ value. C_0 is the bearing static rating load value. If

the bearing model is not decided, it is the way to assume a $\frac{F_a}{C_0}$ value temporarily and then to check if the evaluating C_0 value is right or not after the bearing model has been decided.

2) Select the corresponding e values from the bearing catalog table of e , X , Y .

3) First judge if the F_a/F_r is larger than the e value or not, and then to check out the corresponding X and Y values.

4) Calculate the dynamic equivalent load P from the equation (4-12).

The contact angle of the roller bearing does not change with the axial load, therefore, after selecting the bearing model, the corresponding X , Y , e values can be checked out directly from the bearing catalog.

(2) The axial load calculation of two oppositely mounted angular contact ball bearings or tapered roller bearings Under the radial load and the axial load are subjected on the angular contact ball bearing or on the tapered roller bearing at the same time, the load acting center is not on the intersection point of the bearing width middle line with the shaft center line, but is on the intersection point of the every rolling element load vector with the shaft center line, as shown in Figure 4-3.

In Figure 4-3, two bearings are expressed by bearing I and bearing II, the distances of the load acting centers with the bearing outer ring large

end faces are expressed by a_I and a_{II} .

Under the radial load acting, the angular contact ball bearing and the tapered roller bearing will be producing an inner axial load at the same time intending to separate the bearing inner ring and the bearing outer ring. While two angular contact

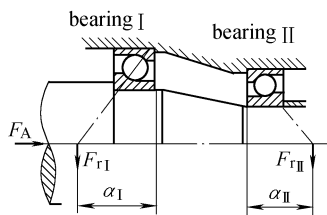


Figure 4-3 Angular contact ball bearing load acting center

ball bearings or two tapered roller bearings are oppositely mounted, one bearing inner axial load will be part of the other bearing outside acting axial load. Therefore, while calculating the dynamic equivalent load P of the angular contact ball bearing and the tapered roller bearing, the bearing inner axial load must be taken into account.

According to the bearing load distribution theory (details in Section 2.3) the bearing inner axial load S are as follows:

For the angular contact ball bearing $S = 1.216 \tan \alpha F_r$

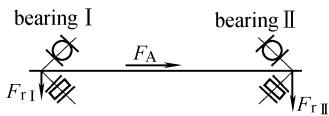
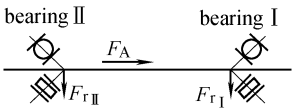
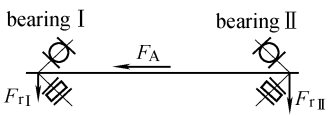
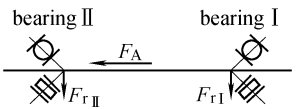
For the tapered roller bearing $S = 1.266 \tan \alpha F_r$

Because the contact angle α of the angular contact ball bearing changes with the axial load F_a , but the judgment factor $e = 1.25 \tan \alpha$. But the contact angle α of the tapered roller bearing does not change with the axial load F_a , its axial factor is as: $Y = 0.4 \cot \alpha$. The values of e and Y can be checked out from the bearing catalog. In order to simplify the calculation, the bearing inner axial load S are as follows:

$$\left. \begin{array}{l} \text{For the angular contact ball bearing} \quad S = e F_r \\ \text{For the tapered roller bearing} \quad S = \frac{F_r}{2Y} \end{array} \right\} \quad (4-13)$$

Table 4-11 lists the axial load calculation equations of the bearing I and the bearing II for different mounting conditions. In Table 4-11, the upper part expresses the angular contact ball bearing symbol and the lower part expresses the tapered roller bearing symbol.

Table 4-11 Axial load calculation equations of the angular contact ball bearing and the tapered roller bearing for different mounting conditions

Mounting symbol	Order	Load condition	F_{aI}	F_{aII}
	1	$S_I \leq S_{II}, F_A \geq 0$	$S_{II} + F_A$	S_{II}
	2	$S_I > S_{II}$ $F_A \geq S_I - S_{II}$	$S_{II} + F_A$	S_{II}
	3	$S_I > S_{II}$ $F_A < S_I - S_{II}$	S_I	$S_I - F_A$
	4	$S_I \geq S_{II}$ $F_A \geq 0$	S_I	$S_I + F_A$
	5	$S_I < S_{II}$ $F_A \geq S_{II} - S_I$	S_I	$S_I + F_A$
	6	$S_I < S_{II}$ $F_A < S_{II} - S_I$	$S_{II} - F_A$	S_{II}

Note: In Table: S_I, S_{II} — bearing I, bearing II inner axial load; F_{aI}, F_{aII} — bearing I, bearing II subjected axial load; F_{rI}, F_{rII} — bearing I, bearing II subjected radial load; F_A — outside acting axial load.

Knowing from the bearing load distribution theory in Section 2.3 that under the radial load acting, the angular contact ball bearing and the tapered roller bearing will be producing an inner axial load at the same time intending to separate the bearing inner ring and the bearing outer ring. The inner axial load S is as:

$$S = \frac{J_a(T)}{J_r(T)} \tan \alpha F_r$$

where S — the inner axial load in a bearing;

$J_a(T)$ — load distribution axial integral can be checked out from Table 2-12;

$J_r(T)$ — load distribution radial integral can be checked out from Table 2-12;

T — load distribution factor;

F_r — radial load subjected on a bearing.

Knowing from the S calculation equation that under different load distribution factor T , the inner axial load is different. All of the outside acting axial load F_a always can be balanced by the inner axial load in a bearing. In order to ensure the normal working for angular contact ball bearing and tapered roller bearing, there should be at least halve of the rolling elements in loading, that is the half-raceway loading, in this time $T = 0.5$.

From Table 2-12, it can be checked out that:

For the angular contact ball bearing: $J_r(0.5) = 0.2288$;

$$J_a(0.5) = 0.2782;$$

For the tapered roller bearing: $J_r(0.5) = 0.2453$;

$$J_a(0.5) = 0.3090;$$

So, it can be got that:

For the angular contact ball bearing: $S = 1.216 \tan \alpha F_r$

For the tapered roller bearing: $S = 1.266 \tan \alpha F_r$

The contacting condition is very complicated in rolling bearing. For ball bearings, all are point contacting, but for roller bearings, some are line contacting, but some are point contacting. In order to simplify the calculation, it is assumed to take the uniform calculation equation as:

$$S = 1.25 \tan \alpha F_r$$

Where, α is the actual bearing contact angle. For tapered roller bearing, the contact angle α designates the rolling element contact angle with the outer ring raceway. These interior structure parameters can not be found from the ordinary rolling bearing reference, but the axial factor Y and the judgment factor e can be checked out from the bearing catalog. Therefore, the inner axial load can be calculated by the following equations:

For the angular contact ball bearing $S = e F_r$

For the tapered roller bearing $S = \frac{F_r}{2Y}$

(3) Dynamic equivalent load calculation under changing working conditions Many machines are working under changing loads, some

machines are working under changing loads and changing speeds. Knowing from the equations (4-1) and (4-2) that for ball bearings, the bearing fatigue life is inversely proportional with the 3 power of the dynamic equivalent load; for roller bearings, the bearing fatigue life is inversely proportional with the 10/3 power of the dynamic equivalent load.

Therefore, for heavy load, although its acting time is very short, its influence upon the bearing fatigue life is large. Thus, for the changing load condition, the average dynamic equivalent load P_m must be used while calculating the bearing fatigue life.

Assuming that after the load P_1 has been acting for N_1 million rotations on the bearing, the bearing load capacity would be consumed for $\frac{N_1}{L_1}$. According to the fatigue accumulation damage principle, if the bearing would be failed after P_2 and P_3 continue acting for N_2 , N_3 million rotations, we can get the following equation:

$$\frac{N_1}{L_1} + \frac{N_2}{L_2} + \frac{N_3}{L_3} + \cdots = 1 \quad (4-14)$$

Where L_1 , L_2 , L_3 — the bearing fatigue lives that should be achieved under acting of P_1 , P_2 , P_3 .

For ball bearing, the following equation could be got from the equation (4-14):

$$L_1 = \left(\frac{C}{P_1}\right)^3; L_2 = \left(\frac{C}{P_2}\right)^3; L_3 = \left(\frac{C}{P_3}\right)^3; \cdots$$

If L expresses the bearing total life, P_m expresses the average dynamic equivalent load, then the following equation could be got:

$$L = \left(\frac{C}{P_m}\right)^3$$

If putting L_1 , L_2 , L_3 into the above equation, the average dynamic equivalent load P_m for ball bearings is as:

$$P_m = \sqrt[3]{\frac{N_1 P_1^3 + N_2 P_2^3 + N_3 P_3^3 + \cdots}{L}} \quad (4-15)$$

For the same reason, the roller bearing average dynamic equivalent load P_m could be as follows:

$$P_m = \sqrt[10/3]{\frac{N_1 P_1^{10/3} + N_2 P_2^{10/3} + N_3 P_3^{10/3} + \cdots}{L}} \quad (4-16)$$

For the changing working conditions shown in Figure 4-4, their average dynamic equivalent load P_m could be got from the equations (4-15) or (4-16). On ordinary condition, the error is very small if taking the exponent as 3 for roller bearings. It is also recommended to take the uniform exponent as 3 for both ball bearings and roller bearings in some reference.

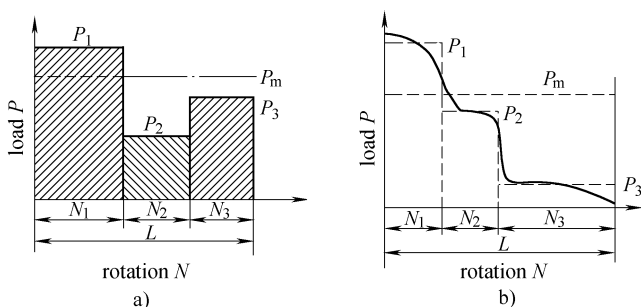


Figure 4-4 Changing working condition

If the dynamic equivalent load changes periodically, then the equations can be rewritten as follows:

For ball bearing
$$P_m = \sqrt[3]{\frac{P_1^3 N_1 + P_2^3 N_2 + P_3^3 N_3 + \cdots}{T}} \quad (4-17)$$

For roller bearing
$$P_m = \sqrt[10/3]{\frac{P_1^{10/3} N_1 + P_2^{10/3} N_2 + P_3^{10/3} N_3 + \cdots}{T}} \quad (4-18)$$

If the bearing runs under the loads P_1 , P_2 , P_3 acting in this turn, the corresponding rotation speed is n_1 , n_2 , n_3 . The ratio of the running time under every working condition to the total running time is a_1 , a_2 , a_3 respectively.

The average dynamic equivalent load is:

For ball bearing
$$P_m = \sqrt[3]{\frac{P_1^3 n_1 a_1 + P_2^3 n_2 a_2 + P_3^3 n_3 a_3 + \cdots}{n_m}} \quad (4-19)$$

For roller bearing
$$P_m = \sqrt[10/3]{\frac{P_1^{10/3} n_1 a_1 + P_2^{10/3} n_2 a_2 + P_3^{10/3} n_3 a_3 + \cdots}{n_m}} \quad (4-20)$$

In the equation, $n_m = n_1 a_1 + n_2 a_2 + n_3 a_3 + \cdots$

If the bearing rotation speed keeps not changed, the dynamic equivalent load changes linearly or near linearly between $P_{\min}$ and $P_{\max}$, as shown in Figure 4-5, then the average dynamic equivalent load is:

$$P_m = \frac{P_{\min} + 2P_{\max}}{3} \quad (4-21)$$

If the bearing loads are the load F_1 of no change in both direction and value (such as rotor weight) and the rotating load F_2 which value is not changed (such as the centrifugal force caused by the

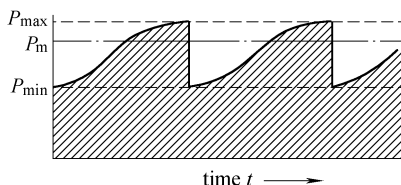
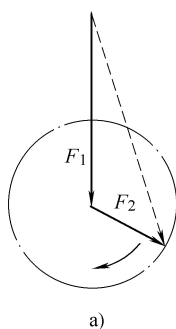


Figure 4-5 Varying load in time Figure

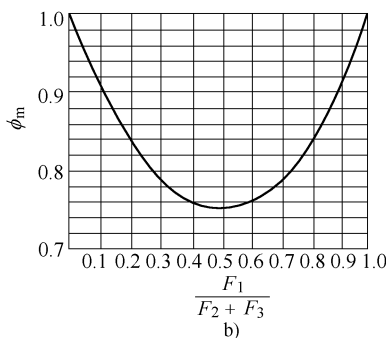
disequilibrium weight) as shown in Figure 4-6a, the average dynamic equivalent load can be calculated from the following equation:

$$F_m = \phi_m (F_1 + F_2) \quad (4-22)$$

The ϕ_m value can be checked out from Figure 4-6b.



a)



b)

Figure 4-6 Load diagram

If the dynamic equivalent load is a function changing with the time, its average dynamic equivalent load P_m can be integrated as follows:

$$\text{For ball bearing: } P_m = \left[\int_0^L \frac{(XF_r + YF_a)^3 dN}{L} \right]^{\frac{1}{3}} \quad (4-23)$$

$$\text{For roller bearing: } P_m = \left[\int_0^L \frac{(XF_r + YF_a)^{\frac{10}{3}} dN}{L} \right]^{\frac{3}{10}} \quad (4-24)$$

For the complicated load changing condition, if the load change could be drawn out as a load diagram, the dynamic equivalent load could be determined by diagram method. The steps are as follows:

- 1) to make the dynamic equivalent load diagram in proper scale.
- 2) To divide the bearing load diagram as many enough small sections, to calculate the dynamic equivalent load P^3 at every section end, and then to draw the P^3 diagram in proper scale.
- 3) To calculate the surrounding area of the P^3 diagram. To calculate the average value of P^3 using the abscissa divided by the surrounding area.
- 4) To calculate the average dynamic equivalent load as follows:

$$P_m = \sqrt[3]{P_{\text{average}}^3}$$

(4) The influence of the load types on the dynamic equivalent load
The loads subjected on bearing can be divided as: object weight supported by bearing; gear, belt transmission force; centrifugal force caused by the disequilibrium weight of rotating rotor and additional load caused by machine vibration and shocking. Generally, the object weight and the gear, belt transmission force can be calculated, but it is very difficult to calculate precisely the additional load caused by machine vibration and shocking. Usually, it is multiplied by an experience coefficient according to the machine working condition. In this case, the dynamic equivalent load is:

$$P = f_p (XF_r + YF_a) \quad (4-25)$$

Where f_p — load type coefficient can be check out from Table 4 – 12.

Table 4 – 12 Load type coefficient

Load character	f_p	Example
no shock force or just little shock force	1.0 ~ 1.2	electric motor, steam turbine, ventilator, water pump
medium shock force	1.2 ~ 1.8	vehicles, machine tool, transmission device, crane, metallurgy equipment, internal combustion engine, reduction gear box
large shock force	1.8 ~ 3.0	crusher, steel rolling mill, petroleum drilling machine, vibrator

4. Life modification factor a_1

The calculated life from the equation (4-1) or (4-2) is associated with 90% reliability, that is the basic rating life L_{10} . That is meant that for a large amount of identical bearings, there would be 90% bearings that can achieve or exceed this basic rating life, but in other words, there would be 10% bearings that possibly have been failed before achieving this basic rating life. For a bearing, this basic rating life is meant that the probability to achieve this life is 90%.

After analysing the bearing test result of 2250 bearings, Tallian verified that under the probability of survival between 40% and 93%, the Lundberg-Palmgren rolling bearing fatigue life theory is conformed with the bearing test data. With the quick development of science and technology, the higher reliability requirement for rolling bearing is more and more. For the reliability higher than 90%, Tallian put forward the values of life modification factor a_1 . The bearing life calculation is:

$$L_{na} = a_1 L_{10} \tag{4-26}$$

Where L_{na} — the bearing life of any reliability requirement, where n expresses the reliability requirement. For example, if the reliability requirement is 99%, then the life is L_{1a} ; if the reliability requirement is 95%, then the life is L_{5a} .

a_1 — reliability modification factor

The a_1 values given by Tallian are listed in Table 4 – 13. This reliability modification factor has been listed in International Standard ISO 281:1990 and China National Standard GB/T6391 — 2003.

Table 4 – 13 Life reliability modification factor a_1

Reliability S	90	95	96	97	98	99	99.2	99.4	99.6	99.8	99.9	99.92	99.94	99.95
L_n	L_{10}	L_5	L_4	L_3	L_2	L_1	$L_{0.8}$	$L_{0.6}$	$L_{0.4}$	$L_{0.2}$	$L_{0.1}$	$L_{0.08}$	$L_{0.06}$	$L_{0.05}$
a_1	1	0.64	0.55	0.47	0.37	0.25	0.22	0.19	0.16	0.12	0.093	0.087	0.080	0.077

Note: The a_1 values in Table 4 – 13 are the modified a_1 values listed in ISO 281:2007. The values have been modified slightly compared with the corresponding values in the previous edition of this International Standard.

Therefore, the rolling bearing life calculation equation for the high

reliability requirement is:

$$\text{For ball bearing:} \quad L_{na} = a_1 \left(\frac{C}{P} \right)^3 \quad (4-27)$$

$$\text{For roller bearing:} \quad L_{na} = a_1 \left(\frac{C}{P} \right)^{\frac{10}{3}} \quad (4-28)$$

5. Other factors influencing bearing fatigue life

(1) temperature

The rolling bearing fatigue life calculation introduced in this chapter is based on the dynamic load rating C . The dynamic load rating C is the bearing load carrying capacity determined at the working temperature condition under 120°C and this temperature designates the measuring temperature at the bearing outer ring. Therefore, if the bearing working temperature is over 120°C , the contact temperature on the rolling element and the raceway would exceed the tempering temperature of the bearing component. In this condition, the bearing component would lose the original dimension stability. Thus, the rolling bearings only can be working under the temperature 120°C .

If the bearing working temperature is higher than the temperature 120°C , it should put forward to the manufacturer to select the bearings that should have been taken through special heat treatment. The bearing component hardness would be reduced under high working temperature. Therefore the bearing dynamic load rating C would be reduced. That is:

$$C_T = f_T C \quad (4-29)$$

Where f_T — temperature factor, listed in Table 4-14;

C_T — bearing dynamic load rating under T ;

C — bearing dynamic load rating in the bearing catalog.

Table 4-14 Temperature factor f_T

bearing working temperature/ $^{\circ}\text{C}$	<120	125	150	175	200	225	250	300
f_T	1	0.95	0.90	0.85	0.80	0.75	0.70	0.60

(2) Bearing floating clearance The rolling bearing dynamic load rating is determined on the condition of the bearing working radial floating

clearance $G_r = 0$. If the bearing working radial floating clearance $G_r > 0$ or $G_r < 0$ (negative floating clearance, on radial preload condition), the bearing load capacity would be influenced.

The rolling bearing fatigue life is largely related to the maximum rolling element load $Q_{\max}$. If $Q_{\max}$ is notably increased, the bearing fatigue life would be notably reduced. Therefore, any factor influencing $Q_{\max}$ would influence the bearing fatigue life. Thus, the bearing radial floating clearance is an important factor influencing $Q_{\max}$.

If using L' expresses the bearing life on the radial floating clearance as G_r , then from the bearing dynamic load rating theory discussed in chapter 2 (Section 2.6), the following equations can be deduced:

$$\text{For ball bearing} \quad \frac{L'}{L} = \left[\frac{J_1 (0.5)}{J_r (0.5)} \frac{J_r (T)}{J_1 (T)} \right]^3 \quad (4-30)$$

$$\text{For roller bearing} \quad \frac{L'}{L} = \left[\frac{J_1 (0.5)}{J_r (0.5)} \frac{J_r (T)}{J_1 (T)} \right]^{\frac{10}{3}} \quad (4-31)$$

In these equations: L — life calculated from the equations (4-1) and (4-2).

$J_r (0.5)$, $J_r (T)$ — load distribution integral can be checked out from Table 2 – 12 in chapter 2;

$J_1 (0.5)$, $J_1 (T)$ — integral can be checked out from Table 2 – 17 in chapter 2.

According to the calculation results, generally, the bearing life is the longest on the condition of the load distribution factor $T = 0.7 \sim 0.8$. In this case $\frac{L'}{L}$ is about 1.1, that is meant that the bearing life is the longest on the condition that the bearing radial floating clearance is little smaller than $G_r = 0$. On the condition that the load distribution factor range is between $T < 0.5$ and $T > 0.8$, the bearing life will be reduced quickly. Special on the condition of negative floating clearance, because of the large heat developed by bearing, the bearing life will be reduced more quickly.

(3) Hardness Usually the hardness of the bearing component is 58 ~ 64 HRC. The rolling bearing dynamic load rating is determined on the

condition of this hardness range. If the actual bearing component hardness is lower than the above range, The rolling bearing dynamic load rating should be reduced, the reducing value should be determined by experiment.

If the hardness of the bearing component is lower than 58 HRC, some reference introduced that the rolling bearing dynamic load rating can be reduced as:

$$C' = C \left(\frac{\text{HRC}}{58} \right)^{3.6} \quad (4-32)$$

Where C' — rolling bearing dynamic load rating under the actual hardness;

C — rolling bearing dynamic load rating checked from the bearing catalog;

HRC — actual hardness in Rockwell hardness.

6. Rolling bearing life calculation examples

[**Example 4 – 1**] It has been decided to select the deep groove ball bearings as the shaft supports in some machinery, knowing that the shaft bearing seat diameter: $d = 35\text{mm}$, the subjecting radial load: $F_r = 1.0\text{kN}$, the axial load: $F_a = 0.85\text{kN}$, the bearing rotating speed: $n = 1800\text{r/min}$. There is a little vibration on bearing during working, the load type factor can select as $f_p = 1.1$. The bearing working temperature is lower than 120°C , the required bearing rating life $L_h > 5000\text{h}$. To select the bearing model.

[**Resolve**] According to the listed deep groove ball bearing C_{0r} value in the bearing catalog, temporarily to select:

$$\frac{F_a}{C_0} = 0.04$$

From Table, the corresponding values with $\frac{F_a}{C_0} = 0.04$ can be checked out as:

$$e = 0.24, X = 0.56, Y = 1.8$$

But
$$\frac{F_a}{F_r} = \frac{0.85}{1.0} = 0.85 > e$$

From the equation (4-25), the dynamic equivalent load can be calculated as:

$$P = f_p (X F_r + Y F_a) = 1.1 * (0.56 * 1.0 + 1.8 * 0.85) \text{ kN} \\ = 2.299 \text{ kN}$$

From Table 4-3 we check out: $f_n = 0.265$ for $n = 1800 \text{ r/min}$.

From Table 4-4 we check out: $f_h = 2.15$ for $L_h = 5000 \text{ h}$.

The required bearing load rating can be calculated out from the equation (4-11):

$$C = \frac{f_h}{f_n} P = \frac{2.15}{0.265} * 2.299 \text{ kN} = 18.65 \text{ kN}$$

From the bearing catalog we check out: the dynamic load rating and the static load rating of the 6207 deep groove ball bearing are close to the required values:

$$C_r = 25.5 \text{ kN} \quad C_{0r} = 15.2 \text{ kN}$$

The rotation speed limit is:

grease lubrication: $n_j = 8500 \text{ r/min}$

oil lubrication: $n_j = 11000 \text{ r/min}$

outside dimension:

$$d = 35 \text{ mm} \quad D = 72 \text{ mm} \quad B = 17 \text{ mm}$$

Therefore, the 6207 deep groove ball bearing can meet the requirement.

$$[\text{Checking}] \quad \frac{F_a}{C_0} = \frac{0.85}{15.2} = 0.0559$$

The factors of deep groove ball bearing can be got from the bearing catalog by the interpolation:

$$X = 0.56 \quad Y = 1.694$$

So the dynamic equivalent load P can be calculated as:

$$P = 1.1 * (0.56 * 1.0 + 0.85 * 1.694) \text{ kN} = 2.199 \text{ kN}$$

The life factor f_h can be calculated out from the equation of Table 4-2 as:

$$f_h = f_n \frac{C}{P} = 0.265 * \frac{25.5}{2.199} = 3.072$$

The bearing life L_h can be checked out from Table 4-4:

$$L_h = 14000 \text{ h} > 5000 \text{ h}$$

The static equivalent load P_0 for $F_a/F_r \leq 0.8$ is:

$$P_{0r} = F_r$$

The static equivalent load P_0 for $F_a/F_r \geq 0.8$ is:

$$P_{0r} = 0.6 F_r + 0.5 F_a$$

$$= 0.6 * 1.0 \text{ kN} + 0.5 * 0.85 \text{ kN} = 1.025 \text{ kN}$$

So, $P_{0r} < C_{0r} = 15.2 \text{ kN}$.

The bearing rotation speed limit is $n_j = 8500 \text{ r/min} > n = 1800 \text{ r/min}$. Therefore, the selected deep groove ball bearing 6207 can meet the requirement.

[**Example 4 - 2**] It has been decided to select two tapered roller bearings for a transmission shaft shown in Figure 4 - 7: $F_{dI} = 17 \text{ kN}$, $F_{dII} = 18 \text{ kN}$, $F_A = 8 \text{ kN}$, $n = 155 \text{ r/min}$. There has medium shocking, the load type factor $f_p = 1.5$. It has been determined to select bearing I as 32018, bearing II as 32022. To calculate the bearing lives.

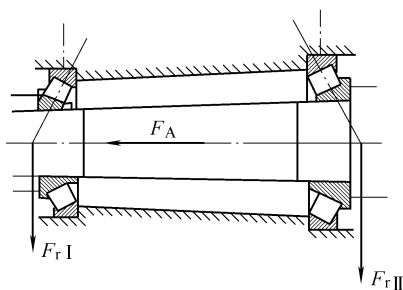


Figure 4 - 7 Support load

[**Resolve**] Every bearing parameter of the bearing I and the bearing II of tapered roller bearings can be checked out from the bearing catalog as listed in Table 4 - 15.

Table 4 - 15 Selecting bearing parameters

Bearing model	C_r/kN	C_{0r}/kN	e	Y	Y_0
I 32018	178	270	0.42	1.4	0.8
II 32022	258	402	0.43	1.4	0.8

The inner axial load S from the equation (4-13) is as follows:

$$S_I = \frac{F_{rI}}{2Y_I} = \frac{17 \text{ kN}}{2 * 1.4} = 6.07 \text{ kN}$$

$$S_{II} = \frac{F_{rII}}{2Y_{II}} = \frac{18 \text{ kN}}{2 * 1.4} = 6.43 \text{ kN}$$

Thus $S_I < S_{II}$; $F_A > S_{II} - S_I$

The inner axial load S and the outside axial load F_A are conformed with the condition of the number 5 in Table 4 - 11.

Therefore $F_{aI} = S_I = 6.07 \text{ kN}$

$$F_{aII} = S_I + F_A = (6.07 + 8) \text{ kN} = 14.07 \text{ kN}$$

$$\text{For bearing I} \quad \frac{F_{aI}}{F_{rI}} = \frac{6.07}{17} = 0.375 < e_I = 0.42$$

$$\text{For bearing II} \quad \frac{F_{aII}}{F_{rII}} = \frac{14.07}{18} = 0.782 > e_{II} = 0.43$$

Checking out from the bearing catalog for tapered roller bearings:

While $F_a/F_r \leq e$, $P_r = F_r$

While $F_a/F_r > e$, $P_r = 0.4F_r + YF_a$

Therefore, the calculated dynamic equivalent load is as:

$$\begin{aligned} P_I &= 17 \text{ kN} * f_F = 17 \text{ kN} * 1.5 = 25.5 \text{ kN} \\ P_{II} &= 0.4F_{rII} + 1.4F_{aII} \\ &= (0.4 * 18 \text{ kN} + 1.4 * 14.07 \text{ kN}) * f_F \\ &= 26.89 \text{ kN} * 1.5 = 40.35 \text{ kN} \end{aligned}$$

Checking out from Table 4-3: while the rotating speed $n = 155 \text{ r/min}$, $f_n = 0.631$.

Calculating out from the equation in Table 4-2:

$$\begin{aligned} f_{hI} &= f_n \frac{C_I}{P_I} = \frac{178}{25.5} * 0.631 = 4.405 \\ f_{hII} &= f_n \frac{C_{II}}{P_{II}} = 0.631 * \frac{258}{40.35} = 4.035 \end{aligned}$$

Checking out from Table 4-4: $L_{hI} = 70000 \text{ h}$

$$L_{hII} = 55000 \text{ h}$$

[**Example 4-3**] A transmission shaft is rotating on the conditions listed in Table 4-16 for high reliability requirement, its working temperature is lower than 120°C . It has been decided to select the cylindrical roller bearing N2320. To calculate the bearing life of 99% reliability.

[**Resolve**] The bearing N2320 working conditions have been listed in Table 4-16.

From the equation (4-20), the average rotating speed and average dynamic equivalent load can be got as:

$$\begin{aligned} n_m &= n_1 a_1 + n_2 a_2 + n_3 a_3 \\ &= (1000 * 0.3 + 500 * 0.5 + 100 * 0.2) \text{ r/min} \\ &= 570 \text{ r/min} \end{aligned}$$

$$P_m = \sqrt[10]{\frac{30^{10/3} * 1000 * 0.3 + 45^{10/3} * 500 * 0.5 + 60^{10/3} * 100 * 0.2}{570}} \text{ kN}$$

$$= 39.84 \text{ kN}$$

Table 4 – 16 Bearing N 2320 working conditions

P/kN	$n/(\text{r/min})$	ratio of the life at working condition to the total life
$P_1 = 30$	$n_1 = 1000$	$a_1 = 30\%$
$P_2 = 45$	$n_2 = 500$	$a_2 = 50\%$
$P_3 = 60$	$n_3 = 100$	$a_3 = 20\%$

Checking out from the bearing catalog, the dynamic load rating of N2320 is as: $C = 435 \text{ kN}$.

Checking out from Table 4 – 13, the 99% reliability factor is: $a_1 = 0.21$.

Checking out from Table 4 – 3, the rotation speed factor related to rotation speed $n_m = 570 \text{ r/min}$: $f_n = 0.427$.

Calculating out from the equations (4-11) and (4-2):

$$f_h = f_n \frac{C}{P_m} = 0.427 * \frac{435}{39.84} = 4.66$$

Checking out from Table 4 – 4: $L_h = 85000 \text{ r/min}$

Calculating out from the equations (4-26): the bearing life of 99% reliability: $L_{na} = a_1 L_{10} = 0.21 * 85000 = 17850 \text{ h}$

4.1.4 Selecting Rolling Bearing According to the Static Load Rating

For bearings subjecting to static load, such as on the static condition and on the low speed condition (designates the bearing of $D_m n < 10^5$, D_m is the bearing average diameter (mm), n is the bearing rotation speed (r/min) and on the slow swing condition, the main failure mode is not the fatigue failure, but is the plastic deformation on the contact place of the rolling element with the raceway, and if the static load is exceedingly large, there would be a dent pit on the contacting surface that would be causing the running accuracy reduction, the friction torque increase and the large vibration and noise. Therefore, in these cases, the bearing selection should

be according to the static load rating.

The basic equation for selecting bearing according to the static load rating is as follows:

$$f_s = \frac{C_0}{P_0} \tag{4-33}$$

- Where C_0 — static load rating (kN) ;
 P_0 — static equivalent load rating, calculated from the equation in Section 2.9 (kN) ;
 f_s — permissible static load factor, the important factor preventing from producing the large plastic deformation on the contacting point of the rolling element.

For the conditions that the bearing rotation speed is very slow and the requirements for the running accuracy and friction torque are not strict, higher contacting stress could be permissible, in this case, $f_s = 1 \sim 1.5$ could be taken; for the conditions that the requirements for the running accuracy and friction torque are more strict, in this case, $f_s > 1.5$ should be taken. The f_s values should be taken referring to Table 4 - 17.

For the thrust aligning roller bearing, it should be taken that $f_s \geq 4$.

Table 4 - 17 Permissible static load factor f_s

Bearing application condition	f_s	
	ball bearing	roller bearing
running requires special low noise	2	3
having shocking, vibration	1.5	2
ordinary running	1	1.5

For higher rotation speed bearing, the bearing model should be selected by the equations (4-1) and (4-2) and then check if the selected bearing static load rating could meet the requirement. For the bearings subjected to shocking load or the bearing life that is very short, it is specially needed to check if the selected bearing static load rating could meet the requirement. For the low speed running bearing (designates the bearing of $D_m n < 10^5$) or the slow swing bearing, the bearing model should be selected by two

methods such as the calculations of the static load rating and contact fatigue life, and then selecting the large dimension one.

When selecting bearing according to the static load rating, we must pay attention to the rigidity of the matching component with the bearing. For the bearing housing with weak rigidity, higher f_s value should be taken. For the bearing housing with strong rigidity, lower f_s values should be taken.

Calculating out the C_0 value from the equation (4-33), the bearing model can be determined from the bearing catalog.

The bearing static load rating would be reduced for the bearings which hardness is lower than HRC 58:

$$C'_0 = \eta_H C_0 \quad (4-34)$$

in which
$$\eta_H = f_H \left(\frac{HV}{800} \right)^2 \leq 1$$

Where C'_0 — the bearing static load rating under the actual hardness

f_H — factor related to the contact type, listed in Table 4-18

HV — Vickers hardness

Table 4-18 Factor f_H

Contact type	f_H
ball contacts with surface (aligning ball bearing)	1
ball contacts with deep groove raceway (deep groove and angular contact ball bearing)	1.5
roller contacts with roller (cylindrical roller bearing)	2
roller contacts with surface	2.5

For some applications, the shaft seat surface and the housing bore surface are directly in contact with the rolling elements used as raceway. In these cases, if the hardness of the shaft seat surface and the housing bore surface can achieve 58 ~ 64 HRC, the ordinary calculation methods for the static load rating and the dynamic load rating can be used. If the hardness of the shaft seat surface and the housing bore surface are lower than the above range value, the static load rating and the dynamic load rating can be got from the equation (4-34).

4.1.5 Rolling Bearing Wear Life

In rolling bearing, there are many kinds of wear mode, such as the adhering wear, the abrasive wear; the corrosion wear and the micro-vibrating wear. On the condition of no good lubrication, it is easily existed that the adhering wear would occur on the relative moving surfaces, and even that the surface burning would occur on the surfaces. If the bearing sealing is not so good that could not prevent the invading of the foreign dust and sand particles, and the lubricant is not clean including some impurity or the bearing cleaning is not very clean leaving some dirt in it, the wear would occur. The corrosion wear would occur if there are moisture and water vapor invading to the bearing. The micro-vibrating wear would occur if the bearing is not rotating and under micro-vibration nearby for a long time.

The wear will make the surfaces of the bearing becoming rough, the bearing radial floating clearance becoming large and the bearing vibration and noise increased, the bearing friction torque increased and at last the bearing would lose the working capacity, even some vital accident can happen.

Usually, the bearing wear life designates the accumulated total rotations or the working hours under certain rotation speed finished by the bearing before losing normal working capacity.

1. Permissible wear coefficient

There is no perfect method for calculating the bearing wear life till now. At present, the only indication for evaluating the wear life is the increased bearing radial floating clearance amount. The increased bearing radial floating clearance amount can be determined according to the bearing type, the bearing dimension and the requirement for the bearing by the machine. The bearing wear life is the bearing working hours before achieving the increased amount of the permissible bearing radial floating clearance. The permissible bearing radial clearance increased amount is the basic data.

Introducing the permissible wear coefficient f_v :

$$f_v = V/e_0 \quad (4-35)$$

Where e_0 — wear rate (μm) related to the bearing inner bore diameter, the value can be determined from the equation (4-36), or from Figure 4-8;

V — bearing radial floating clearance increased amount (μm).

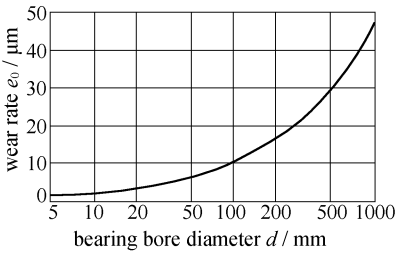


Figure 4-8 Bearing wear rate e_0 value

$$e_0 = 0.46d^{0.67} \tag{4-36}$$

Where d — bearing inner bore diameter (mm).

According to the bearing working condition, we select the wear coefficient region and the wear coefficient f_v from Table 4-19. The low value of the permissible wear coefficient region in Table 4-19 is suitable for the serious working condition, that is the smaller permissible wear amount. According to the wear coefficient f_v and the wear coefficient region, the bearing wear life can be determined from Figure 4-9. In Figure 4-9, the relationships of the wear coefficient f_v to the bearing wear life are expressed by 10 regions divided by a group of curves. Every region expresses the working condition and the wearing situation, the direction from a to k expresses the gradually worse working condition and the intensifying wearing.

Table 4-19 Wear coefficient region and permissible wear coefficient f_v

Bearing position		Wear coefficient region	Wear coefficient f_v
automobile	gear	$g \sim k$	5 ~ 8
	transmission shaft	$h \sim k$	3 ~ 6
	water pump	k	5 ~ 7
	clutch	k	5 ~ 7
	wheel hub	$h \sim j$	4 ~ 6

(continued)

Bearing position		Wear coefficient region	Wear coefficient f_v
motor	electric motor	$i \sim k$	3 ~ 5
	standard motor	$c \sim d$	3 ~ 5
	big motor	$b \sim d$	3 ~ 5
	main transmission motor	$c \sim d$	3 ~ 5
machine tool	lathe spindle	$a \sim b$	0.5 ~ 1.5
	milling spindle	$a \sim b$	0.5 ~ 1.5
	drilling spindle	$b \sim c$	1 ~ 2
	grinding spindle	$c \sim d$	0.5 ~ 1
machine tool	precision grading spindle	$c \sim d$	0.5
	presser flywheel	$d \sim f$	3 ~ 8
	presser crankshaft	$d \sim e$	3 ~ 5
	electric instrument	$g \sim h$	3 ~ 8
	pneumatic instrument	$g \sim h$	3 ~ 8
gear	ordinary gear	$d \sim e$	3 ~ 8
	large gear	$c \sim d$	6 ~ 10
railroad vehicle	passenger vehicle	$c \sim d$	8 ~ 12
	good train	$c \sim d$	8 ~ 12
	locomotive	$d \sim e$	6 ~ 10
transmission equipment	mine well belt driving	$c \sim d$	5 ~ 10
	belt transmission	$g \sim k$	10 ~ 20
	supporting roll		
	belt wheel	$e \sim f$	10 ~ 15
	excavator drive shaft	$c \sim e$	5 ~ 10
wind machine	small wind machine	$f \sim h$	5 ~ 8
	middle wind machine	$d \sim f$	3 ~ 5
	large wind machine	$c \sim d$	3 ~ 5
pump	centrifugal pump	$d \sim f$	3 ~ 5
	compressor	$d \sim f$	3 ~ 5
metallurgy machine	crusher	$f \sim g$	8 ~ 10
	milling roll	$e \sim f$	6 ~ 10
	vibration screen	$e \sim f$	4 ~ 6
	tube milling	$f \sim g$	12 ~ 18

(continued)

Bearing position		Wear coefficient region	Wear coefficient f_v
paper machine	wet part	$b \sim c$	7 ~ 10
	dry part	$a \sim b$	10 ~ 15
	precise machine	$b \sim c$	5 ~ 8
	polishing machine	$a \sim b$	4 ~ 8
wood machine	planing machine	$a \sim b$	1.5 ~ 3.0
	sawing machine	$e \sim g$	3 ~ 4
	plastic machine	$e \sim f$	3 ~ 5
	textile machine	$a \sim e$	2 ~ 8
	centrifugal machine	$e \sim f$	8 ~ 12
	printing machine	$a \sim b$	3 ~ 4

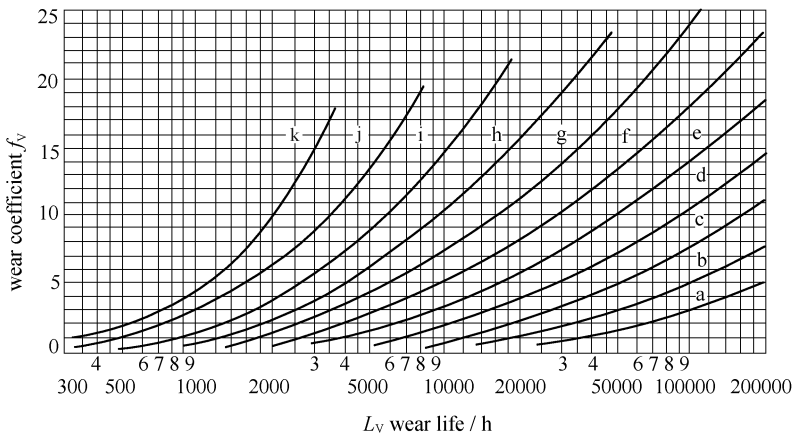


Figure 4-9 Relationship of the wear coefficient f_v to the bearing wear life

2. Wear life calculation

If knowing the permissible increased amount of the bearing radial floating clearance, the bearing wear life could be calculated from the following steps:

- 1) To get the wear rate e_0 from the equation (4-36) or Figure 4-8 according to the bearing bore diameter d .
- 2) To calculate the permissible wear coefficient f_v from the equation (4-35).

3) To select the wear rate region from Table 4 – 19 according to the bearing position.

4) To take the intersection points of the horizontal line of the constant value of the permissible wear coefficient f_v with the corresponding region curve ends, that is the maximum bearing wear life and the minimum bearing wear life.

If having selected the permissible wear coefficient f_v and getting the wear rate e_0 from the equation (4-36) or Figure 4 – 8 according to the bearing bore diameter d , then the increased amount of the bearing radial floating clearance while arriving the bearing wear failure can be calculated from the equation (4-35).

In the above calculation, the only roughly considered factor is the effect of the bearing application condition. Many factors such as the subjecting load, rotating speed, lubrication and running temperature are not precisely calculated, therefore the calculating result only is an approximate value. For the important application conditions, it is recommended to carry on special test for checking.

4.2 Bearing Load Calculation

4.2.1 Load calculation Transferred by Some Common Transmission Components

The loading on rolling bearing supports is usually related to the bearing through varied transmission components. For convenience to the readers, the calculation equations of gear transmission and belt transmission will be introduced as follows.

1. spur cylindrical gear

Figure 4 – 10 is the diagram of the spur cylindrical gear transmission.

In the diagram, the gear I expresses the driving gear expressed by the angle note 1, the gear II expresses the driven gear expressed by the angle note 2 (same as the other gear transmission).

For the circumference force: $F_t = 19098.6 \frac{N}{nd}$

For the centripetal force: $F_s = F_t \tan \alpha$

In which N — transmission power (kW);

n — gear rotation speed (r/min);

d — gear pitch diameter (mm);

α — gear pressure angle ($^\circ$).

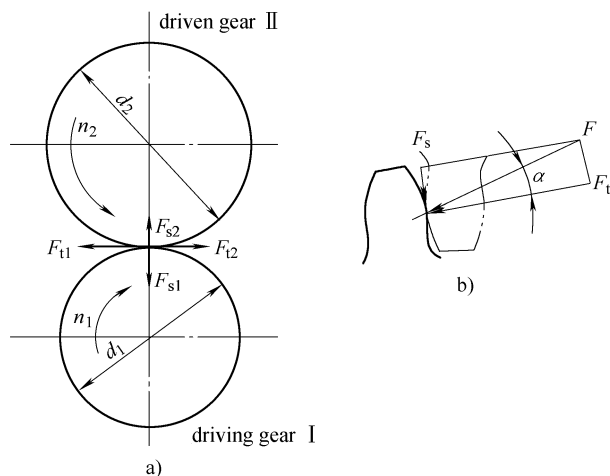


Figure 4-10 Spur cylindrical gear transmission

2. bevel cylindrical gear transmission

Figure 4-11 is the diagram of the bevel cylindrical gear transmission. the transmitted load calculation equations are as follows:

For the circumference force: $F_t = 19098.6 \frac{N}{nd_s}$

For the centrifugal force: $F_s = F_t \frac{\tan \alpha_n}{\cos \beta}$

For the axial force: $F_A = F_t \tan \beta$

The modifying bevel cylindrical gear end face pitch circle diameter is related to the dividing circle diameter:

$$d_s = \frac{d_{0s} \cos \alpha_0}{\cos \alpha}$$

Where d_s — gear end face pitch diameter (mm);

d_{0s} — gear end face dividing circle diameter (mm);

- α_0 — gear end face pressure angle ($^\circ$);
 α — gear end face gearing angle ($^\circ$);
 α_n — gear gearing angle in the normal plane ($^\circ$);
 β_0 — gear spiral angle in dividing circle ($^\circ$);
 N — transmission power (kW);
 n — gear rotation speed (r/min).

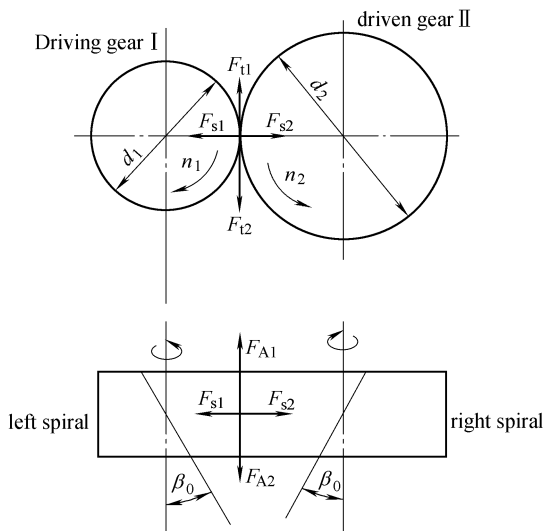


Figure 4-11 bevel cylindrical gear transmission

The principle for determining the axial force direction of the bevel cylindrical gear:

1) When the driving gear has right spiral teeth, the right hand four finger direction expresses the driving gear rotation direction and then the thumb direction expresses the subjecting axial force direction of the driving gear.

2) When the driving gear has left spiral teeth, the left hand four finger direction expresses the driven gear rotation direction and then the thumb direction expresses the subjecting axial force direction of the driven gear. The driven gear axial force direction is in the opposite direction with the driving gear axial force direction.

3. Spur bevel gear transmission

The acting forces of the spur bevel gear transmission are shown in Figure 4-12.

For the circumference force: $F_t = 19098.6 \frac{N}{nd_m}$

For the centrifugal force: $F_{s1} = F_{t1} \tan \alpha_0 \cos \delta_1 = -F_{A2}$

For the axial force: $F_{A1} = F_{t1} \tan \alpha_0 \sin \delta_1 = -F_{s2}$

$$\cos \delta_1 = \frac{i}{\sqrt{i^2 + 1}}$$

Here δ_1 — smaller gear half-taper angle ($^\circ$);

i — transmission ratio;

α_0 — gear standard pressure angle ($^\circ$);

d_m — average pitch circle diameter of spur bevel gear (mm).

4. Helical bevel gear transmission or arc bevel gear transmission

The acting forces of the helical bevel gear transmission are shown in Figure 4-13.

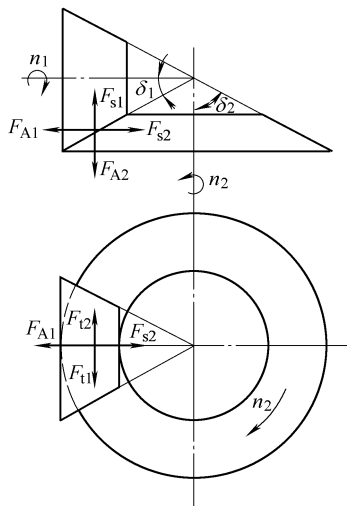


Figure 4-12 Spur bevel gear transmission

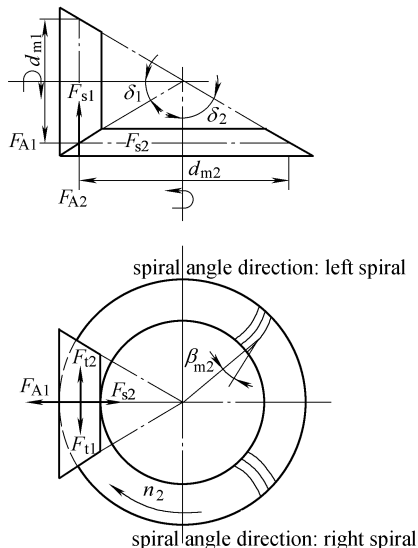


Figure 4-13 Helical bevel gear transmission

The circumference force: $F_t = 19098.6 \frac{N}{nd_m}$

The calculation equations of the centripetal force and the axial force are listed in Table 4 – 20.

When calculating the centripetal force and the axial force, the first thing is to determine the spiral angle direction and the gear rotation direction.

The method to determine the spiral angle direction is as shown in Figure 4 – 14. Facing to the smaller gear end, if the gear teeth length direction leaves the center in clockwise, the spiral angle direction is right spiral; if the gear teeth length direction leaves the center in anticlockwise, the spiral angle direction is left spiral.

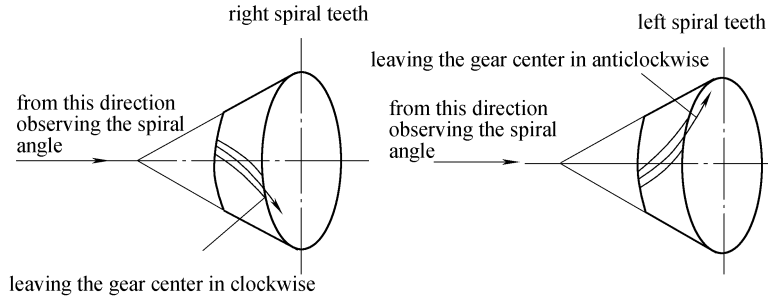


Figure 4 – 14 Determine spiral angle direction principle

The method to determine the gear rotation direction is as shown in Figure 4 – 15. Facing to the larger gear end, we can tell if the gear rotation direction is in clockwise or the gear rotation direction is in anticlockwise.

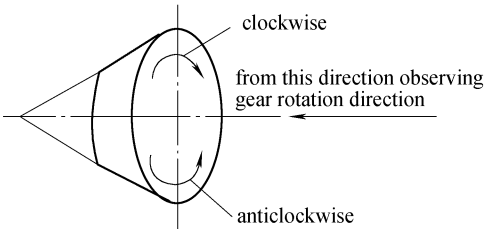


Figure 4 – 15 Determine gear rotation direction principle

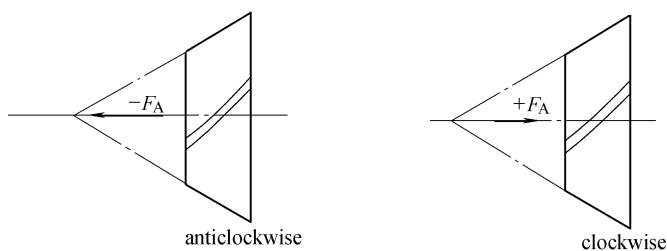
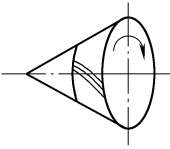
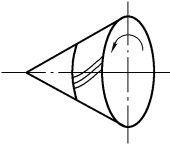
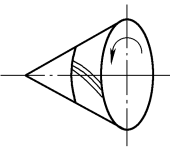
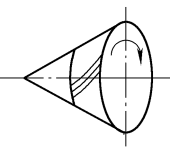


Figure 4-16 Determine arc bevel gear axial force direction principle

Table 4-20 Bevel gear axial force F_A calculation equation

Condition		Calculation equation
driving gear right teeth clockwise running 	or driving gear left teeth anticlockwise running 	$F_{A1} = F_{s2} = F_t \left(\tan \alpha_n \frac{\sin \delta_1}{\cos \beta_m} - \tan \beta_m \cos \delta_1 \right)$ $F_{A2} = F_{s1} = F_t \left(\tan \alpha_n \frac{\sin \delta_2}{\cos \beta_m} + \tan \beta_m \cos \delta_2 \right)$
driving gear right teeth anticlockwise running 	or driving gear left teeth clockwise running 	$F_{A1} = F_{s2} = F_t \left(\tan \alpha_n \frac{\sin \delta_1}{\cos \beta_m} + \tan \beta_m \cos \delta_1 \right)$ $F_{A2} = F_{s1} = F_t \left(\tan \alpha_n \frac{\sin \delta_2}{\cos \beta_m} - \tan \beta_m \cos \delta_2 \right)$

Note: α_n — gear gearing angle in the normal plane ($^\circ$);

β_m — gear width middle point spiral angle in ($^\circ$);

δ_1 — gear half-taper angle ($^\circ$).

In Table 4-20, if let:

$$a_1 = \tan \alpha_n \frac{\sin \delta_1}{\cos \beta_m}$$

$$b_1 = \tan \alpha_n \frac{\cos \delta_1}{\cos \beta_m}$$

$$a_2 = \tan \alpha_n \frac{\sin \delta_2}{\cos \beta_m}$$

$$b_2 = \tan \alpha_n \frac{\cos \delta_2}{\cos \beta_m}$$

When the driving gear right teeth clockwise running (or when the driving gear left teeth anticlockwise running)

$$F_{A1} = F_{s2} = F_t (a_1 - b_1)$$

$$F_{A2} = F_{s1} = F_t (a_2 + b_2)$$

When the driving gear right teeth anticlockwise running (or when the driving gear left teeth clockwise running)

$$F_{A1} = F_{s2} = F_t (a_1 + b_1)$$

$$F_{A2} = F_{s1} = F_t (a_2 - b_2)$$

According to the given values of α_n , δ , β_m , the a_1 , a_2 , b_1 , b_2 values can be calculated out from the above equations, and then the axial force and the centripetal force can be calculated out.

5. Worm transmission

The acting forces of the worm transmission are as shown in Figure 4 - 17.

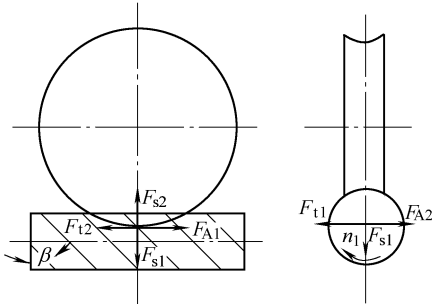


Figure 4 - 17 Worm transmission acting forces

In the worm transmission, the worm circumference force F_{t1} is in the opposite direction with the worm wheel axial force F_{A2} , but their values are the same. The worm axial force F_{A1} is in the opposite direction with the worm wheel circumference force F_{t2} , but their values are the same.

$$F_{t1} = F_{A2} = 19098.6 \frac{N_1}{d_1 n_1}$$

$$F_{t2} = F_{A1} = 19098.6 \frac{N_1 \eta}{d_2 n_2}$$

$$F_{s1} = F_{s2} = F_{t2} \frac{\tan \alpha_{0n}}{\cos \beta}$$

Where F_{t1} — worm circumference force (kN);
 F_{t2} — worm wheel circumference force (kN);
 F_{A1} — worm axial force (kN);
 F_{A2} — worm wheel axial force (kN);
 F_{s1} — centrifugal force (kN);
 F_{s2} — worm wheel centripetal force (kN);
 N_1 — worm transmission power (kW);
 η — transmission efficiency;
 α_{0n} — pressure angle in the normal plane;
 β — worm helix angle;
 d_1 — worm pitch circle diameter (mm);
 d_2 — worm wheel pitch circle diameter (mm);
 n_1 — worm rotation speed (r/min);
 n_2 — worm wheel rotation speed (r/min).

6. Belt transmission

The acting force F_t in the belt transmission can be got from the following equation:

$$F_t = 19098.6 \frac{N}{dn}$$

Where N — transmission power (kW);
 d — belt wheel diameter (mm);
 n — belt wheel rotation speed (r/min).

The transmission force F_t is the difference of the belt tighten end tension force and the belt loose end tension force.

If using S_1 expresses the belt tighten end tension force and S_2 expresses the belt loose end tension force, then $F_t = S_1 - S_2$ as shown in Figure 4-18.

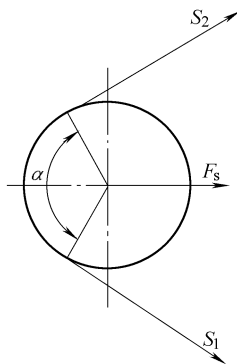


Figure 4-18 Belt transmission acting forces

Therefore , the centripetal force F_s that the belt acts on the shaft is related to the initial tension force , the included angle and the rotation speed. Usually , the approximate calculation of the centripetal force F_s that the belt acts on the shaft can be got by the transmission force F_t time an experienced factor f_b :

$$F_s = f_b F_t$$

Where F_t — belt transmission force ;
 F_s — force that the belt acts on the shaft ;
 f_b — belt factor , listed in Table 4 - 21.

Table 4 - 21 Belt factor f_b

Belt type	f_b
V-leather belt	2 ~ 2.5
flat leather belt with tension pulley	2.5 ~ 3.5
flat leather belt without tension pulley, silk belt	3.5 ~ 4.5
single layer leather belt, rubber belt	4 ~ 5
double layer leather belt, cotton belt, hemp belt	5 ~ 6
chain	1.25 ~ 1.5

Note: in Table 4 - 21 , larger values are appropriate when center distance is short and the belt speed is low.

4.2.2 Load Calculation of Double Support Shaft Bearing

1. Statically determinate solution

In ordinary machine, most of them are the double support shafts. The loading on a rolling bearing support is usually transmitted to the bearing outer ring through the shaft on which the bearing is mounted. Sometimes, however, the loading is transmitted to the bearing outer ring through the housing that encompasses the bearing outer ring, such as a wheel bearing. In most applications, we consider the double support shaft as a statically determinate system. The support reaction force can be calculated from the system force equilibrium equation. The support reaction force can be used as the rolling bearing acting loads to calculate the bearing rating life.

The non-aligning rolling bearing can carry some bending moment to make the bearing inner ring angular misaligned relative to the bearing outer ring. If considering the influence of the bending moment for the shaft deformation, the number of the support reaction forces will be more than the number of the equilibrium equations, therefore, the shaft system can be as a statically indeterminate system.

In most bearing applications, we consider the double support shaft as a statically determinate system for precisely determining the bearing subjecting load. In this condition, it is assumed that there is no influence of bearing deformation considering the support system as a simple support system, and the bearings are only subjected to the radial load perpendicular to the shaft axis. But in important bearing application, such as in high speed machine application and in machine tool spindle application, in order to determine the shaft deflection, the shaft dynamic load and the bearing loading condition, it must be considered that the support system is a statically indeterminate system.

Table 4 - 22 lists the simple calculation equations of the support reaction force for some typical conditions. Because the value of the support reaction force is equal to the value of the bearing load and the direction is in the opposite direction, therefore, these equations can be used for calculating the bearing load.

In Table 4 - 22 F_{rI} — bearing I radial load (kN) ;

F_{rII} — bearing II radial load (kN) ;

F_A — axial load (kN) ;

F_t — gear transmission circumference force (kN) ;

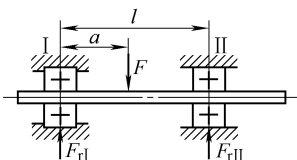
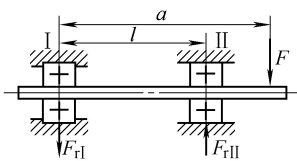
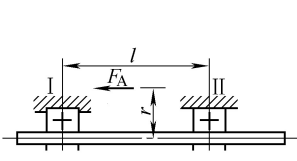
F_s — gear transmission centripetal force (kN) ;

d — gear pitch circle diameter (mm) ;

r — gear pitch circle radius (mm).

According to the shaft support structure, we determine which bearing is subjected to the axial load.

Table 4 - 22 Double support shaft reaction force equations

order	sketch	F_{rI}	F_{rII}	description
1		$\frac{F(l-a)}{l}$	$\frac{Fa}{l}$	
2		$\frac{F(l-a)}{l}$	$\frac{Fa}{l}$	load F is on the outside of the double supports, $(l-a)$ is negative, thus F_{rI} is negative.
3		$\frac{F_A r}{l}$	$-\frac{F_A r}{l}$	If F_A direction is in opposite with the sketched direction, using $-F_A$.

(continued)

order	sketch	F_{rI}	F_{rII}	description
4		$\frac{1}{l} \sum_1^n [F_i(l - a_i) + F_{Ai}r_i]$	$\frac{1}{l} \sum_1^n [F_i a_i - F_{Ai}r_i]$	If F_{Ai} direction is in opposite with the sketched direction, using $-F_{Ai}$. If no axial load, then taking $F_{Ai} = 0$.
5		$\frac{1}{l} \int_{a_0}^l (l - x) q(x) dx$	$\frac{1}{l} \int_{a_0}^l x q(x) dx$	
6		$F_{rI} = \frac{1}{l} \left(\sqrt{[F_t(l - a)]^2 + \left[F_s(l - a) \pm \frac{F_A d}{2}\right]^2} \right)$ $F_{rII} = \frac{1}{l} \left(\sqrt{(F_t a)^2 + \left(F_s a \mp \frac{F_A d}{2}\right)^2} \right)$		If the reaction forces caused by F_s and F_A are in the same direction, taking the positive sign. In opposite, taking the negative sign. $\otimes$ expresses the F_t direction that is perpendicular to the paper plane.

2. Statically indeterminate solution

If considering the bearing reaction bending moment, the number of the equilibrium equations will be smaller than the number of the reaction forces, therefore, the shaft system would be as a statically indeterminate system. In order to get the bearing load, the shaft deformation condition must be added.

Knowing from the material mechanics about the beam deformation, the relationship of the deflection y , turning angle θ and the bending moment are as follows:

$$EJy'' = M(x) \quad (4-37)$$

$$\theta = y' \quad (4-38)$$

Here $y'' = \frac{d^2 y}{dx^2}$;

$$y' = \frac{dy}{dx};$$

θ — shaft turning angle;

$M(x)$ — subjected bending moment on the shaft, the bending moment, the principle for determining the sign of the bending moment as shown in Figure 4-19;

E — shaft material Modulus of elasticity;

J — shaft section inertia moment.



Figure 4-19 Principle for determining the sign of the bending moment

For solid shaft :

$$J = \frac{\pi d^4}{64}$$

For hollow shaft:

$$J = \frac{\pi}{64} (d_e^4 - d_i^4)$$

d is the shaft diameter, d_e , d_i are the shaft outside diameter and inner

diameter of the hollow shaft.

Figure 4-20 is the condition that a concentrated load is supported between two bearings.

It can be considered to be divided as two parts.

The first part: $0 \leq x \leq a$

$$M(x) = F_{r1}x + M_1$$

From the equation 4-37,

the following equation can be got:

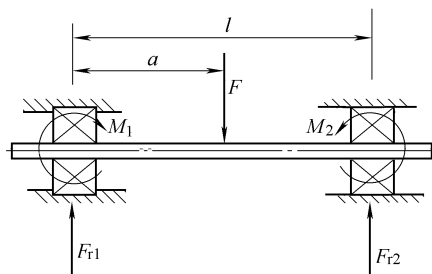


Figure 4-20 Double support sketch

$$EJy'' = F_{r1}x + M_1$$

$$EJy' = \int (F_{r1}x + M_1) dx = F_{r1} \frac{x^2}{2} + M_1x + C_1 \quad (4-39)$$

$$\begin{aligned} EJy &= \int \left(F_{r1} \frac{x^2}{2} + M_1x + C_1 \right) dx \\ &= F_{r1} \frac{x^3}{6} + M_1 \frac{x^2}{2} + C_1x + D_1 \end{aligned} \quad (4-40)$$

Assuming that the boundary condition is:

$$x=0 \quad y' = \theta_1 \quad y = \delta_{r1}$$

Put the boundary condition into the equation (4-39) and equation (4-40), then the values can be got:

$$C_1 = EJ\theta_1$$

$$D_1 = EJ\delta_{r1}$$

Then the equation (4-39) and equation (4-40) can be rewritten as:

$$EJy' = F_{r1} \frac{x^2}{2} + M_1x + EJ\theta_1 \quad (4-41)$$

$$EJy = F_{r1} \frac{x^3}{6} + M_1 \frac{x^2}{2} + EJ\theta_1x + EJ\delta_{r1} \quad (4-42)$$

The second part: $a \leq x \leq l$

$$M(x) = F_{r1}x - F(x-a) + M_1$$

From the equation (4-37) the following equations can be got:

$$EJy'' = F_{r1}x - F(x-a) + M_1$$

$$EJy' = F_{r1} \frac{x^2}{2} - F \frac{(x-a)^2}{2} + M_1 x + C_2 \quad (4-43)$$

$$EJy = F_{r1} \frac{x^3}{6} - F \frac{(x-a)^3}{6} + M_1 \frac{x^2}{2} + C_2 x + D_2 \quad (4-44)$$

Assuming that the boundary condition is:

$$x = l \quad y' = \theta_2 \quad y = \delta_{i2}$$

Put the boundary condition into the equation (4-43) and equation (4-44), then the values can be got:

$$C_2 = -F_{r1} \frac{l^2}{2} + F \frac{(l-a)^2}{2} - M_1 l + EJ\theta_2$$

$$D_2 = F_{r1} \frac{l^3}{3} - \frac{F}{6} (l-a)^2 (2l+a) + M_1 \frac{l^2}{2} - EJ\theta_2 l + EJ\delta_{i2}$$

Put C_2 , D_2 into the equation (4-43) and equation (4-44), then the following equations can be got:

$$EJy' = \frac{F_{r1}}{2}(x^2 - l^2) + \frac{F}{2}[(l-a)^2 - (x-a)^2] + M_1(x-l) + EJ\theta_2 \quad (4-45)$$

$$EJy = \frac{F_{r1}}{6}(x^3 - 3l^2x + 2l^3) + \frac{F}{6}[(l-a)^2(3x-2l-a) - (x-a)^3] + \frac{M_1}{2}(x-l)^2 + EJ\delta_{i2} + EJ\theta_2(x-l) \quad (4-46)$$

According to the continuing condition, in $x = a$, the calculated value from the equations (4-41) and (4-42) should be equal; the calculated value from the equations (4-42) and (4-46) should be equal; thus the following equations can be got:

$$F_{r1} \frac{l^2}{2} + M_1 l = F \frac{(l-a)^2}{2} + EJ(\theta_2 - \theta_1) \quad (4-47)$$

$$\begin{aligned} & F_{r1} \frac{l^2(3a-2l)}{6} + M_1 \left(\frac{2al-l^2}{2} \right) \\ &= F \frac{[(l-a)^2(a-l)]}{3} \\ &+ EJ[\delta_{i2} - \delta_{r1} - \theta_1 a + \theta_2(a-l)] \end{aligned} \quad (4-48)$$

From the equations (4-47) and (4-48) the following equations can be got:

$$F_{r1} = F \frac{(l-a)^2(2a+l)}{l^3} + \frac{6EJ}{l^2} \left[\theta_1 + \theta_2 + \frac{2(\delta_{r1} - \delta_{r2})}{l} \right] \quad (4-49)$$

$$M_1 = -F \frac{(l-a)^2 a}{l^2} - \frac{2EJ}{l} \left[2\theta_1 + \theta_2 + \frac{3(\delta_{r1} - \delta_{r2})}{l} \right] \quad (4-50)$$

From the force equilibrium equation: $\sum F = 0$ and $\sum M = 0$, the following equations can be got:

$$F_{r2} = F \frac{a^2(3l-2a)}{l^3} - \frac{6EJ}{l^2} \left[\theta_1 + \theta_2 + \frac{2(\delta_{r1} - \delta_{r2})}{l} \right] \quad (4-51)$$

$$M_2 = -F \frac{(l-a)a^2}{l^2} + \frac{2EJ}{l} \left[\theta_1 + 2\theta_2 + \frac{3(\delta_{r1} - \delta_{r2})}{l} \right] \quad (4-52)$$

Equations (4-49) and (4-50) are the general solution of the equation for considering the reaction moment of the double support shaft. If considering the double support shaft as a statically determinate solution, then, assuming that the radial deflection is the same and the reaction moment is zero, that is:

$$\delta_{r1} - \delta_{r2} = 0$$

$$M_1 = M_2 = 0$$

If put this condition into the equations (4-49) ~ (4-52), the following equations can be got:

$$F_{r1} = \frac{F(l-a)}{l}$$

$$F_{r2} = \frac{Fa}{l}$$

This is the same as the result of the order 1 in Table 4-22.

4.2.3 Load Calculation of Three Support Shaft Bearing

1. General solution method

The three support shaft is a statically indeterminate solution. If the shaft deflection is large, the bearing load can be calculated by using the shaft deformation equations. Usually, the three support shaft shown in Figure 4-21a can be simplified as two double support shafts shown in Figure 4-21b. And then solving these double support shafts by the above section.

In Figure 4-21:

$$F_{r2} = F'_{r2} + F''_{r2} \quad (4-53)$$

$$M_2 = M'_2 - M''_2 \quad (4-54)$$

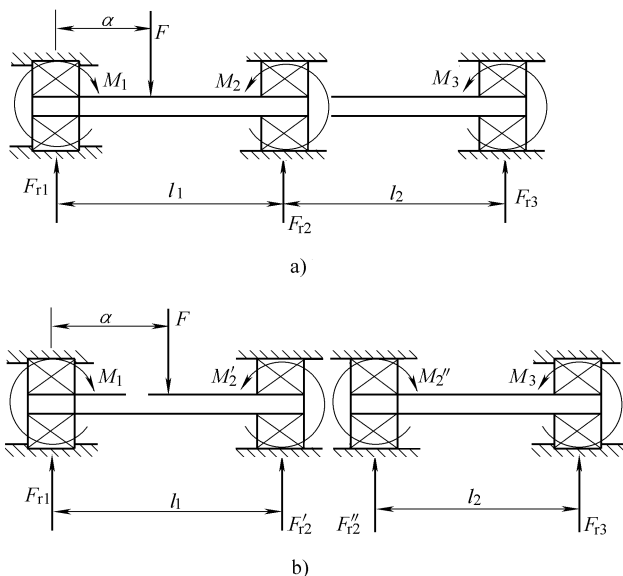


Figure 4-21 Three support shaft simplified solving method

From the equations (4-49) to (4-54) the following equations can be got:

$$F_{r1} = F \frac{(l_1 - a)^2 (2a + l_1)}{l_1^3} + \frac{6EJ_1}{l_1^2} \left[\theta_1 + \theta_2 + \frac{2(\delta_{r1} - \delta_{r2})}{l_1} \right] \quad (4-55)$$

$$\begin{aligned}
F_{r2} = F_{r2}' + F_{r2}'' = F \frac{a^2(3l_1 - 2a)}{l_1^3} \\
- 6E \left[\frac{J_1}{l_1^2}(\theta_1 + \theta_2) - \frac{J_2}{l_2^2}(\theta_2 + \theta_3) \right] \\
- 12E \left[\frac{J_1}{l_1^3}(\delta_{r1} - \delta_{r2}) - \frac{J_2}{l_2^3}(\delta_{r2} - \delta_{r3}) \right]
\end{aligned} \quad (4-56)$$

$$F_{r3} = -\frac{6EJ_2}{l_2^2} \left[\theta_1 + \theta_3 + \frac{2(\delta_{r2} - \delta_{r3})}{l_2} \right] \quad (4-57)$$

$$\begin{aligned}
M_1 = -F \frac{(l_1 - a)^2 a}{l_1^2} \\
- \frac{2EJ_1}{l_1} \left[2\theta_1 + \theta_2 + \frac{3(\delta_{r1} - \delta_{r2})}{l_1} \right]
\end{aligned} \quad (4-58)$$

$$\begin{aligned}
M_2 = M_2' - M_2'' = -F \frac{(l_1 - a) a^2}{l_1^2} \\
+ 2E \left[\frac{J_1}{l_1} (\theta_1 + 2\theta_2) + \frac{J_2}{l_2} (2\theta_2 + \theta_3) \right] \\
+ 6E \left[\frac{J_1}{l_1^2} (\delta_{r1} - \delta_{r2}) + \frac{J_2}{l_2^2} (\delta_{r2} - \delta_{r3}) \right]
\end{aligned} \quad (4-59)$$

$$M_3 = \frac{2EJ_2}{l_2} \left[\theta_2 + 2\theta_3 + \frac{3(\delta_{r2} - \delta_{r3})}{l_2} \right] \quad (4-60)$$

Generally, it is very difficult to determine the θ_1 , θ_2 , θ_3 , δ_{r1} , δ_{r2} , δ_{r3} in the equations (4-55) to (4-60), therefore, it is very complicated to solve the above six unknown values. In order to simply the calculation, some assumptions have to be made to be sufficient enough to precisely determine the three support shaft bearing loads.

2. Simplifying solving method

If assuming that:

1) The bearing reaction bending moment on every support are as zero;
 2) The radial deflections on every bearing are very little and can be omitted;

3) The shaft section inertia moment is the same.

The following equations can be got:

$$\delta_{r1} - \delta_{r2} = 0$$

$$\delta_{r2} - \delta_{r3} = 0$$

$$M_1 = M_2 = M_3 = 0 \quad J_1 = J_2$$

Putting these into the equations (4-58) ~ (4-60) can get:

$$2\theta_1 + \theta_2 = -F \frac{(l_1 - a)^2 a}{2EJl_1} \quad (4-61)$$

$$\frac{\theta_1 + 2\theta_2}{l_1} + \frac{2\theta_2 + \theta_3}{l_2} = F \frac{(l_1 - a)a^2}{2EJl_1^2} \quad (4-62)$$

$$\theta_2 + 2\theta_3 = 0 \quad (4-63)$$

From the equations (4-61) ~ (4-63), we get θ_1 , θ_2 , θ_3 , and then again put into the equations (4-55) ~ (4-57), the following equations can be got:

$$F_{r1} = \frac{F(l_1 - a)[2l_1(l_1 + l_2) - a(l_1 + a)]}{2l_1^2(l_1 + l_2)} \quad (4-64)$$

$$F_{r2} = \frac{Fa(l_1^2 - a^2 + 2l_1l_2)}{2l_1^2l_2} \quad (4-65)$$

$$F_{r3} = -\frac{F(l_1^2 - a^2)a}{2l_1l_2(l_1 + l_2)} \quad (4-66)$$

Under the above assumptions, the bearing load calculation equations for different load conditions are listed in Table 4-23.

3. Bearing load calculation for few shaft deflection

If the shaft rigidity is very large and the shaft deflection can be omitted, then the shaft position under loads is as shown in Figure 4-22. In Figure 4-22, δ_{r1} , δ_{r2} , δ_{r3} express the bearing radial deflection respectively.

From the theory about the bearing deflection and the load distribution, the following equations can be got:

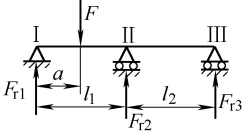
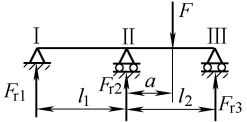
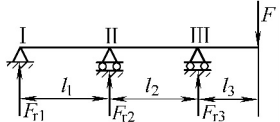
$$\delta_{r1} = K'_1 F_{r1}^n \quad (4-67)$$

$$\delta_{r2} = K'_2 F_{r2}^n \quad (4-68)$$

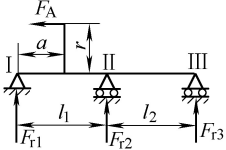
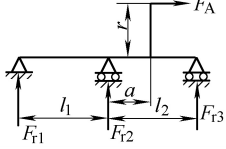
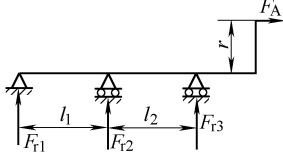
$$\delta_{r3} = K'_3 F_{r3}^n \quad (4-69)$$

Here, for ball bearing $n = \frac{2}{3}$

Table 4-23 Three support shaft bearing load calculation equation

Sketch	Shaft bending moment on middle support	F_{r1}	F_{r2}	F_{r3}
	$-\frac{Fa(l_1^2 - a^2)}{2l_1(l_1 + l_2)}$	$\frac{F(l_1 - a)}{2l_1^2(l_1 + l_2)} \\ * [2l_1(l_1 + l_2) - a(l_1 + a)]$	$\frac{Fa(l_1^2 - a^2 + 2l_1l_2)}{2l_1^2l_2}$	$-\frac{F(l_1^2 - a^2)a}{2l_1l_2(l_1 + l_2)}$
	$-\frac{F(l_2 - a)a(2l_2 - a)}{2l_2(l_1 + l_2)}$	$-\frac{Fa(l_2 - a)}{2l_1l_2(l_1 + l_2)} \\ * (2l_2 - a)$	$\frac{F(l_2 - a)}{2l_1l_2^2} \\ * [a(2l_2 - a) + 2l_1l_2]$	$\frac{Fa}{2l_2^2(l_1 + l_2)} \\ * (2l_1l_2 + 3l_2a - a^2)$
	$\frac{Fl_2l_3}{2(l_1 + l_2)}$	$\frac{Fl_2l_3}{2l_1(l_1 + l_2)}$	$-\frac{Fl_3(2l_1 + l_2)}{2l_1l_2}$	$\frac{F(l_2 + l_3)}{l_2} \\ + \frac{Fl_2l_3}{2l_2(l_1 + l_2)}$

(continued)

Sketch	Shaft bending moment on middle support	F_{r1}	F_{r2}	F_{r3}
	$-\frac{F_A r (3a^2 - l_1^2)}{2l_1 (l_1 + l_2)}$	$\frac{F_A r}{2l_1^2 (l_1 + l_2)} \\ * (3l_1^2 + 2l_1 l_2 - 3a^2)$	$\frac{F_A r}{2l_1^2 l_2} (3a^2 - l_1^2 - 2l_1 l_2)$	$-\frac{F_A r (3a^2 - l_1^2)}{2l_1 l_2 (l_1 + l_2)}$
	$-\frac{F_A r (3a^2 + 2l_2^2 - 6al_2)}{2l_2 (l_1 + l_2)}$	$-\frac{F_A r}{2l_1 l_2 (l_1 + l_2)} \\ * (3a^2 + 2l_2^2 - 6al_2)$	$\frac{F_A r}{2l_1 l_2^2} \\ * (3a^2 + 2l_2^2 - 6al_2 - 2l_1 l_2)$	$\frac{F_A r}{2l_2^2 (l_1 + l_2)} \\ * (6al_2 - 3a^2 + 2l_1 l_2)$
	$\frac{F_A r l_2}{2 (l_1 + l_2)}$	$\frac{F_A r l_2}{2l_1 (l_1 + l_2)}$	$-\frac{F_A r (2l_1 + l_2)}{2l_1 l_2}$	$\frac{F_A r (2l_1 + 3l_2)}{2l_2 (l_1 + l_2)}$

Note: 1. The directions of F and F_A are the positive signs if in the same direction as shown in the figure; if in the opposite direction with the direction as shown in the figure, it should put in the $(-F)$ and $(-F_A)$.

2. The directions of F_{r1} , F_{r2} , F_{r3} are the positive signs if in the same direction as shown in the figure.

for roller bearing

$$n = 0.9$$

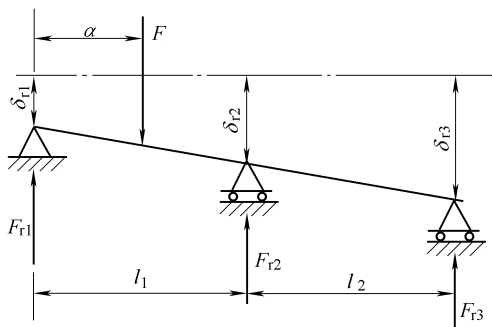


Figure 4-22 Bearing radial deflection sketch

Because the shaft deflection can be omitted, from the geometric relation, the following equation can be got:

$$\frac{\delta_{r2} - \delta_{r1}}{l_1} = \frac{\delta_{r3} - \delta_{r2}}{l_2} \quad (4-70)$$

put the equations (4-67) ~ (4-69) into the equation (4-70):

$$K_2' F_{r2}^n \left(1 + \frac{l_1}{l_2} \right) - K_1' F_{r1}^n = \frac{l_1}{l_2} K_3' F_{r3}^n \quad (4-71)$$

Where K_1' , K_2' , K_3' are the deformation constants that can be got from Section 2.2. If selecting the identical model bearing, then $K_1' = K_2' = K_3'$, in this case the equation (4-71) can be rewritten as:

$$F_{r2}^n \left(1 + \frac{l_1}{l_2} \right) - F_{r1}^n = \frac{l_1}{l_2} F_{r3}^n \quad (4-72)$$

Therefore, the values of F_1 , F_2 , F_3 can be got from the deformation condition and the force equilibrium equation.

4. Bearing load calculation of the support structure as shown in Figure 4-23

Figure 4-23 shown support structure is a typical support structure, on which two identical angular thrust bearings mounted on one support and one radial bearing that can only be subjected to radial load mounted on another support. In this support structure, the shaft rigidity is very large, so, the only considering factor is the bearing deformation. In this case, the bearing

loads can be calculated by the following method.

Two identical angular thrust bearings mounted in face to face or back to back can be considered as a symmetric double row angular thrust bearing. Knowing from Section 2.3 that the load value of every row is related to the relative value of the axial load F_a and the radial F_r .

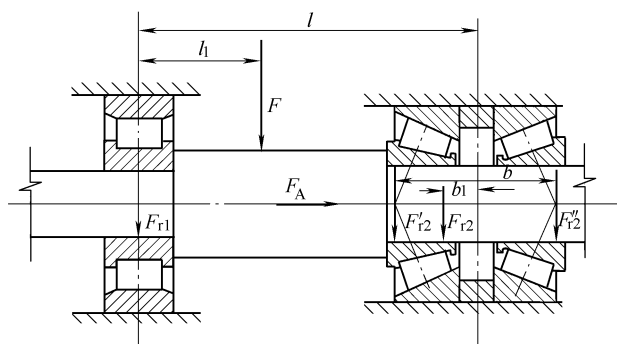


Figure 4-23 Support structure sketch

In Figure 4-23:

F'_{r2} — left angular thrust bearing subjected to radial load;

F''_{r2} — right angular thrust bearing subjected to radial load;

F_{r2} — the combined load of F'_{r2} and F''_{r2} . That is the resultant radial load acting on the double row angular thrust bearing.

b — the distance between two angular thrust bearing load acting centers;

b_1 — the distance between the F_{r2} acting point and the two bearing geometric center;

F_A — axial load applied from outside, because the left support can not subject to the axial load, that is the axial load applied on the double row angular thrust bearing;

F — radial load applied from outside.

From the relationship in Figure 4-23, the following equations can be got:

$$F_{r2} = F'_{r2} + F''_{r2} \quad (4-73)$$

$$\frac{F_{r2}''}{F_{r2}'} = \frac{1 - 2 \frac{b_1}{b}}{1 + 2 \frac{b_1}{b}}$$

(4-74)

The b_1/b value can be got from the the double row angular thrust bearing load distribution theory discussed in Section 2.3 and listed in Table 4-24 and shown in Figure 4-24.

The steps to determine the bearing load by using Figure 4-24 and Table 4-24 are as follows:

Table 4-24 Relationship of b_1/b with $F_A \cot \alpha / F_{r2}$

Ball bearing			Roller bearing		
$F_A \cot \alpha / F_{r2}$	b_1/b	F_{r2}''/F_{r2}'	$F_A \cot \alpha / F_{r2}$	b_1/b	F_{r2}''/F_{r2}'
0	0	1	0	0	1
0.488	0.177	0.477	0.418	0.137	0.510
0.914	0.325	0.212	0.826	0.265	0.306
1.25	0.428	0.078	1.22	0.375	0.142
1.49	0.483	0.017	1.58	0.458	0.043
1.67	0.5	0	1.91	0.5	0

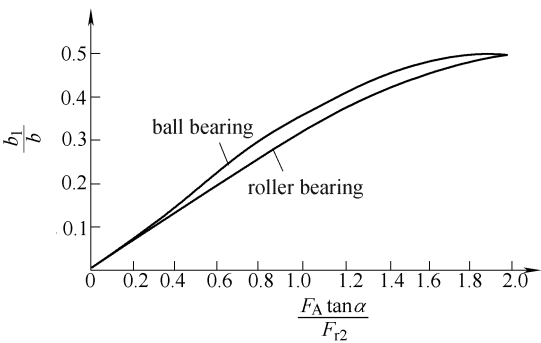


Figure 4-24 Relationship of b_1/b with $F_A \cot \alpha / F_{r2}$

1) Assuming that the resultant load F_{r2} is acted on the geometric center of two angular thrust contact bearing to calculate the first approximate $F_{r2}^{(1)}$ value, that is $b_1^{(1)} = 0$.

2) To calculate the following value:

$$\frac{F_A \cot \alpha}{F_{r2}^{(1)}}$$

In which α is the contact angle of the angular thrust contact bearing.

For angular thrust bearing:

$$\cot \alpha = \frac{1.25}{e}$$

For tapered roller bearing:

$$\cot \alpha = 2.5 Y$$

The e and Y values can be checked out from the rolling bearing catalog.

3) To check out the second approximate $\frac{b_1}{b}$ value related to $\frac{F_A \cot \alpha}{F_{r2}^{(1)}}$

from Figure 4 – 24.

4) To select the proper $b_1^{(2)}$ value as the F_{r2} acting point according to the loading condition.

Through several time approximate calculation, the enough precise F_{r2} value can be got.

For two identical angular thrust bearings mounted in face to face or back to back on one support, their life can be calculated according to the double angular thrust bearing. In this case, for the bearing dynamic rating load and the equivalent dynamic load, the value of the double row angular thrust bearing should be taken. If having known the single row bearing dynamic rating load as C_1 , then the double row bearing dynamic rating load is as:

For angular thrust ball bearing

$$C = 1.62 C_1$$

For tapered roller bearing

$$C = 1.71 C_1$$

4.3 Rolling Bearing Preload

4.3.1 General Introduction

The rolling bearing preload designates the case in that by taking some measure, the bearings produce certain deformation in advance between the rolling elements and the raceways of the inner rings and the outer rings in the bearings after mounting the bearings into the bearing housing and the shaft for keeping that the bearing inner rings and the outer rings are located in the pressed tight condition. The purposes of the rolling bearing preload are increasing support rigidity, reducing vibration and noise, preventing the the sliding of the rolling elements relative to the raceways of the inner ring and the outer ring caused by the inertia moment.

In order to guarantee the smoothly running of the machine tool and the accuracy of the machining parts, it is always required that the machine tool must have large rigidity, that is required that under loading the spindle deformation must be as little as possible. In some transmission devices, in order to guarantee the bevel gear proper engagement, it is always also required that the transmission shaft support must have the large rigidity. These requirements for the support rigidity can be carried out through the preload method.

The rigidity designates the load value required for the deflection per unit. The following equation can expresses the rigidity:

$$k = \frac{dF}{d\delta}$$

Here F — load acting on the bearing;

δ — the bearing deflection under the load F (mm).

The k value is much larger, thus the bearing rigidity is much larger. Knowing from Section 2.2 that the deflection in a ball bearing is in proportional to the $2/3$ power of the load. Therefore, the rigidity is non-linear, so, the rigidity of a ball bearing can not be taken as a constant value. The deflection in a roller bearing is in proportional to the 0.9 power of the load, so, the rigidity of a roller bearing can approximately be taken as

a constant value.

4.3.2 Principle of Rolling Bearing Axial Preload

Knowing from Section 2.2 that the relation equation of the axial deflection with the axial load for an angular contact ball bearing is as follows:

$$\delta_a = K_a F_a^{2/3}$$

Where δ_a — the relative axial displacement amount of the inner ring and outer ring under the load F_a , that is the axial deflection of an angular contact ball bearing (mm);

F_a — axial load acting on the bearing;

K_a — bearing elastic deflection constant. For an angular contact ball bearing, K_a is not a constant, because the actual contact angle α is changeable with the axial load.

According to the above equation, the typical plot of bearing deflection δ_a — F_a is as shown in Figure 4 - 25. Knowing from Figure 4 - 25 that without the bearing preload, for a single bearing, under F_a acting, the bearing axial deflection is as δ_{a1} .

But with the bearing preload F_{a0} , for a single bearing, under the same F_a acting, the bearing axial deflection is as δ_{a2} . It is very obvious that $\delta_{a2} < \delta_{a1}$, therefore, the angular contact ball bearing rigidity could be increased through preloading.

Roller bearing deflection is almost linear with respect to the load, as shown in the following equation:

$$\delta_a = K_a F_a^{0.9}$$

Therefore, there is not much advantageous to be gained by axially

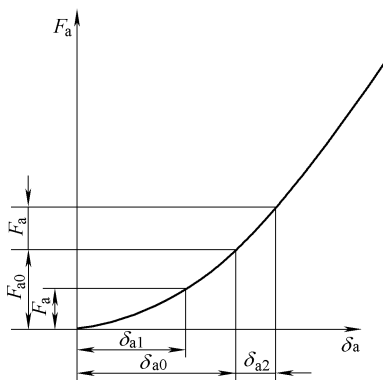


Figure 4 - 25 Single angular contact bearing preload principle

preloading tapered or spherical roller bearings. The single tapered roller bearing rigidity could not be increased through preloading.

If the mounting of two angular contact bearing support system, is in face to face or back to back by preloading method, their support system rigidity could be increased through preloading. According to the preloading method, the preload can be divided as the fixed position preload and the fixed pressure preload.

(1) Fixed position preload

Figure 4-26 shows the fixed position preloads. The distance between two bearings is kept not changed in the fixed position preload. Figure 4-27 shows the load - deflection curve of two angular contact bearings mounted in face to face and back to back in the fixed position preload. The abscissa expresses the axial deflection δ_a , the ordinate expresses the axial load F_a . The intersection point of two bearing load-deflection curves expresses that the preload is F_{a0} and the two bearing deflections both are δ_{a0} .

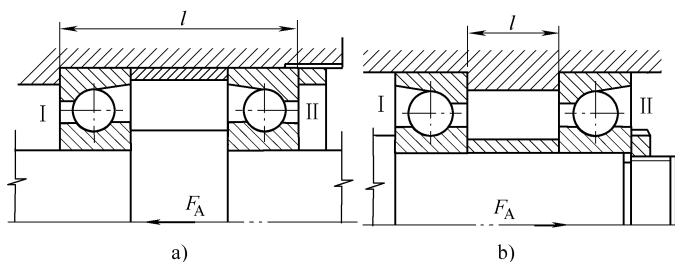


Figure 4-26 Fix position preload structure sketch

a) Face to face b) Back to back

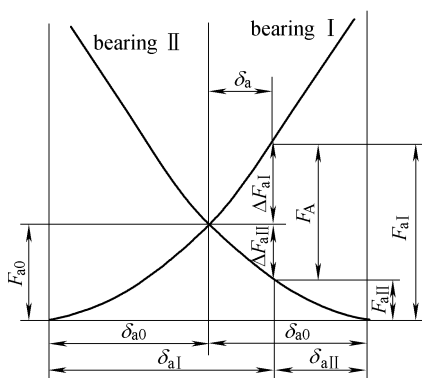
F_A is the axial load applied from the outside on the shaft and assuming that the larger load is carried by the bearing I and the smaller load is carried by the bearing II.

While there is no axial load F_A applied from the outside, the axial loads carried by the bearing I and the bearing II both are F_{a0} and the preload deflection amounts of the bearing I and the bearing II both are δ_{a0} .

While there is an axial load F_A applied from the outside on the shaft,

the load conditions of the bearing I and the bearing II both have changed.

It is supposed that the direction of the axial load F_A is in the direction of increasing the bearing I load (that is the direction shown in Figure 4 - 26). In this case, under the axial load F_A , the shaft has moved δ_a distance along the F direction, correspondingly, the



**Figure 4 - 27 Fix position
preload principle**

deflection amount of the bearing I has increased δ_a , but the deflection amount of the bearing II has decreased δ_a .

Now, it can be seen from Figure 4 - 27:

$$\begin{cases} \delta_{aI} = \delta_{a0} + \delta_a \\ \delta_{aII} = \delta_{a0} - \delta_a \end{cases}$$

At this time, the axial load on the bearing I has increased for ΔF_{aI} , but the axial load on the bearing II has decreased for ΔF_{aII} . That are as follows:

$$\begin{cases} F_{aI} = F_{a0} + \Delta F_{aI} \\ F_{aII} = F_{a0} - \Delta F_{aII} \end{cases}$$

From the force equilibrium of the forces applied on the shaft, the following equation can be got:

$$F_A = F_{aI} - F_{aII}$$

If the axial load F_A increases to the value to let $\Delta F_{aII} = F_{a0}$, the shaft moving amount along the F_A direction is $\delta_a = \delta_{a0}$. It is obviously that, in this case, the bearing II is not subjected to any load at all. Therefore:

$$\begin{cases} \delta_{aI} = 2\delta_{a0} \\ \delta_{aII} = 0 \end{cases}$$

It can be seen very obviously from Figure 4-27 that if the bearing I and the bearing II are not preloaded, under the same axial load F_A , the support system axial deflection is as $2\delta_{a0}$. If the bearing I and the bearing II are preloaded, under the same axial load F_A , the support system axial deflection is only δ_{a0} . Therefore, the support system rigidity can be increased through preloading.

The load letting the bearing II not subjected to any load is called as “unloading load”, expressed by F_{Ax} . From the above derivation, the following equation can be got:

$$F_{Ax} = 2^{3/2} F_{a0} = 2.83 F_{a0}$$

If the axial load F_A is larger than the above value, the axial load is totally carried by the bearing I.

(2) Fixed pressure preload

In the fixed pressure preload, the preloading load is applied by spring. Because the preloading load is not changed with the application of the axial load F_A , this preload method is called as the fixed pressure preload. Figure 4-28 shows the fixed pressure preload principle.

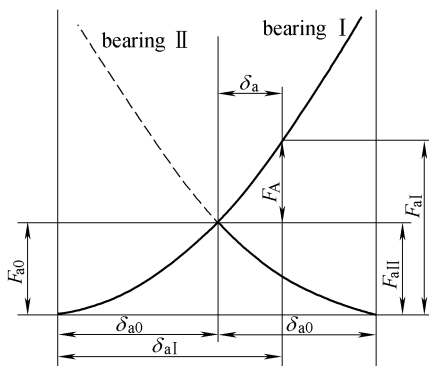


Figure 4-28 Fixed pressure preload principle

Under the preloading load F_{a0} , the preloading deflection amounts of the bearing I and the bearing II both are δ_{a0} . While there is an axial load F_A subjected on the shaft, the shaft has moved δ_a along the F_A direction. The bearing II outer ring presses the inner ring consistently tight by the spring. Because the spring rigidity is much smaller comparing with the bearing rigidity, it is considered approximately that the preloading load of the bearing II is not changed, but the axial load F_A is carried by the bearing I. Thus, the system rigidity increase is not very clearly. Figure

4-29 shows the fixed pressure preload sketch.

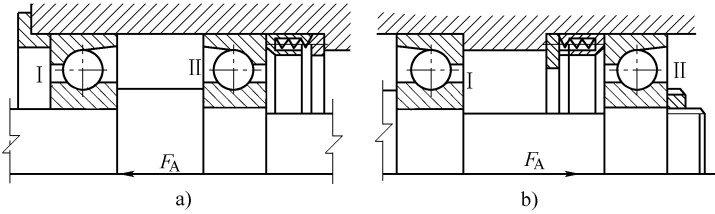


Figure 4-29 Fixed pressure preload structure sketch
a) Face to face b) Back to back

(3) Tapered roller bearing preload

As having mentioned before, the tapered roller bearing deflection is almost linear with respect to the load.

Thus, the single tapered roller bearing system rigidity could not be increased through the preloading method.

But for the paired mounting of tapered roller bearings, the support system rigidity could be increased through the preloading method. Figure 4-30 shows the principle.

Under the preloading load F_{a0} , the preloading deflection amounts of the bearing I and the bearing II both are δ_{a0} . While there is an axial load F_A applied on the shaft, the shaft has moved δ_a along the F_A

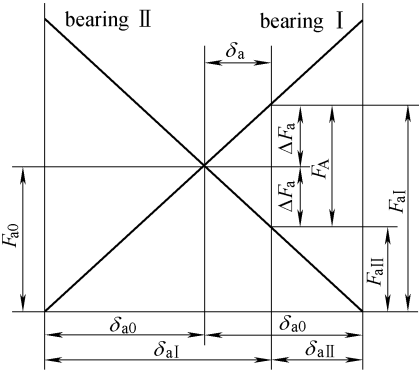


Figure 4-30 Tapered roller bearing preload principle

direction. In this case, the deflection amount of the bearing I is increased for δ_a , but the deflection amount of the bearing II is decreased for δ_a , that is:

$$\delta_{aI} = \delta_{a0} + \delta_a$$

Because the deflection is almost linear with respect to the load, the following equation can be got:

$$\delta_{a0} = K_a \cdot F_{a0}$$

$$\delta_{a1} = K_a \left(F_{a0} + \frac{F_A}{2} \right)$$

Therefore

$$\delta_a = K_a \cdot \frac{F_A}{2}$$

If the bearing I and the bearing II are not preloaded, it is very obviously that the axial load F_A is totally carried by the bearing I. At this time, the shaft movement amount along the F_A direction is δ'_a , that is the deflection amount of the bearing I:

$$\delta'_a = K_a \cdot F_A$$

comparing with the axial deflection amount δ_a of the bearing I in preloading condition, the following relation can be got:

$$\delta'_a = 2\delta_a$$

Therefore, for the paired mounting of tapered roller bearings, the support system rigidity could be increased for one time through the preloading method.

The load letting the bearing II not subjected to any load is called as “unloading load”, expressed by F_{Ax} . From the above derivation, the following equation can be got:

$$F_{Ax} = 2F_{a0}$$

4.3.3 Calculation Equation of Rolling Bearing Axial Deflection

For the standard design of rolling bearing, under the axial load F_a , the relative axial deflection amount calculation equation of the inner ring and the outer ring can be got from the equation (4-55), equation (4-57) and Table 4-8.

For angular contact ball bearing:

$$\delta_a = \frac{0.002}{\sin \alpha} \sqrt[3]{\left(\frac{Q}{9.80665} \right)^2 \cdot \frac{1}{D_w}} = \frac{0.000436}{D_w^{1/3} z^{2/3} \sin^{5/3} \alpha} F_a^{2/3}$$

For tapered roller bearing:

$$\delta_a = \frac{0.0006 Q^{0.9}}{9.80665^{0.9} \sin \alpha L_{we}^{0.8}} = \frac{0.0000768}{L_{we}^{0.8} z^{0.9} \sin^{1.9} \alpha} F_a^{0.9}$$

where Q — rolling element load (N);

- α — bearing actual contact angle;
 D_w — rolling element diameter (mm);
 L_{we} — effective contact length;
 z — rolling element number.

4.3.4 Determination of the Minimum Axial Preload

The bearing preload value should be determined according to the load condition and the application requirement.

If the main purpose of the preload is for reducing the support system vibration and improving the system running accuracy, light preload value should be selected. If the main purpose of the preload is for raising the support system rigidity, heavy preload value should be selected. But if the preload value is overdue large, the friction in the bearing would be increased, the temperature would be raised, the bearing life would be reduced.

Under the pure axial load, while

$$F_A = 2.83 F_{a0} \quad (\text{angular contact ball bearing})$$

$$F_A = 2 F_{a0} \quad (\text{tapered roller bearing})$$

The bearing II is not subjected to any load, unloading the preloading load. The minimum axial preload should be as follows:

$$F_{a0} \geq 0.35 F_A \quad (\text{angular contact ball bearing})$$

$$F_{a0} \geq 0.5 F_A \quad (\text{tapered roller bearing})$$

The inner axial load should be also considered if the radial load and the axial load are applied at the same time.

According to the bearing load distribution theory, in order to increase the bearing rigidity, the number of the loading rolling elements should be as many as possible. The preloading load under which every rolling element in the bearing could subject to the load should be as:

$$\left. \begin{aligned} F_{a0} &\geq 1.7 F_{rI} \cdot \tan \alpha_1 - 0.5 F_A \\ F_{a0} &\geq 1.7 F_{rII} \cdot \tan \alpha_2 + 0.5 F_A \end{aligned} \right\} \quad (\text{angular contact ball bearing})$$

$$\left. \begin{aligned} F_{a0} &\geq 1.9 F_{rI} \cdot \tan \alpha_1 - 0.5 F_A \\ F_{a0} &\geq 1.9 F_{rII} \cdot \tan \alpha_2 + 0.5 F_A \end{aligned} \right\} \quad (\text{tapered roller bearing})$$

4.3.5 Means to Carry Out Axial Preload

(1) Matched pair angular contact ball bearing

Matched pair angular contact ball bearing is a kind of pair preloading bearings, in which the bearing preload amount has been considered during their manufacturing process by grinding off certain deflection amount δ_{a0} on the end faces of two bearing inner rings and two outer rings in advance.

When mounting these two bearings onto the bearing support, it is just required to press the corresponding bearing end faces tightly by the fixing device to get the preloading condition, as shown in Figure 4-31.

There are several kinds of matched pair angular contact ball bearings that having been produced in China:

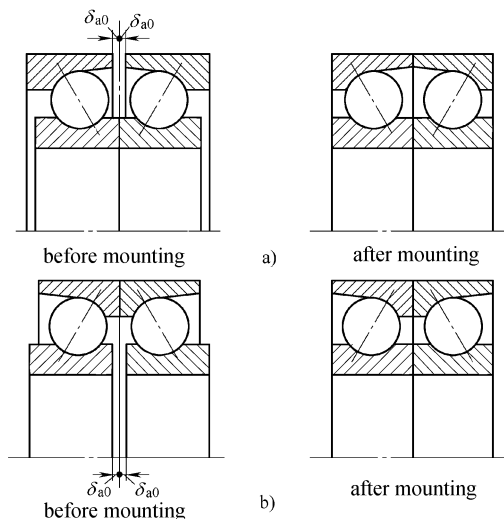


Figure 4-31 Matched pair angular contact ball bearings preloading mounting

a) face to face b) back to back

1) Back to back arrangement matched pair angular contact ball bearings (outer ring wide end faces opposite each other): 70000C/DB type, 70000B/DB type, 70000AC/DB type;

2) Face to face arrangement matched pair angular contact ball bearings

(outer ring narrow end faces opposite each other): 70000C/DF type, 70000AC/DF type, 70000B/DF type;

3) Tandem arrangement matched pair angular contact ball bearings (outer ring wide end faces opposite each other): 70000C/DT type, 70000AC/DT type, 70000B /DT type.

(2) Using shim or spacer

When paired mounting of two angular contact ball bearings, different width shims can be placed between the inner rings and the outer rings of the two bearings in order to get the bearing preloading deflection, as shown in Figure 4-32. The different preloading amount can be got by changing the width of the shims.

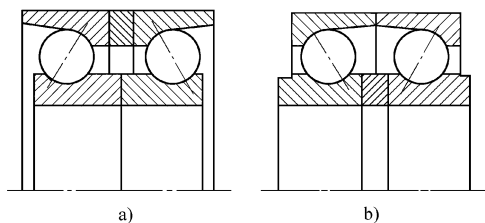


Figure 4-32 Preloading by shims

a) back to back b) face to face

When paired mounting of two angular contact ball bearings within certain distance, different length spacers (sleeves) can be placed between the inner rings and the outer rings of the two bearings in order to get the bearing preloading deflection.

Before axial preloading by using spacer, we should first measure the distances between the inner ring end faces and the outer ring end faces of two bearings.

According to the end face distances and the required preload amount value, the spacer length can be determined.

The end face distances can also be determined by the actual measurement.

(3) Preloading by spring

When mounting the bearings into the support, the preloading load is carried by the spring against the non-rotating rings all the time to get the

fixed pressure preload as shown in Figure 4-29.

This preloading method is very reliable, and has been got comprehensive usage.

4.3.6 Axial Preload Values

There are some special manufacturers manufacturing the paired mounting contact ball bearings in China (such as HRB bearing plant), and the standard of paired mounting two identical angular contact ball bearings has been made: JB/T10186 — 2000 group matched angular contact ball bearing technical condition.

According to the value of the preload, there are three kinds of axial preload, such as light preload, moderate preload and heavy preload. Their values are listed in Table 4-25. In Table 4-25, P_L designates light preload, P_M designates moderate preload, P_H designates heavy preload.

Table 4-25 Paired mounting contact ball bearing preload value (N)

d/mm	7000C			7200C			7300C		
	P_L	P_M	P_H	P_L	P_M	P_H	P_L	P_M	P_H
10	25	50	100	50	100	200	70	140	280
12	25	50	100	60	120	240	80	160	320
15	30	60	120	70	140	280	95	190	380
17	35	70	140	90	180	360	130	260	520
20	50	100	200	115	230	460	145	290	580
25	60	120	240	130	260	520	215	430	860
30	80	160	320	180	360	720	265	530	1060
35	150	300	600	250	500	1000	345	690	1380
40	155	310	620	280	560	1120	405	810	1620
45	190	380	760	310	620	1240	495	990	1980
50	200	400	800	330	660	1320	580	1160	2320
55	270	540	1080	410	820	1640	710	1420	2840
60	280	560	1120	490	980	1960	815	1630	3260
65	280	560	1120	515	1030	2060	925	1850	3700
70	350	700	1400	560	1120	2240	1040	2080	4160
75	360	720	1440	640	1280	2560	1130	2260	4520

(continued)

<i>d</i> /mm	7000C			7200C			7300C		
	<i>P_L</i>	<i>P_M</i>	<i>P_H</i>	<i>P_L</i>	<i>P_M</i>	<i>P_H</i>	<i>P_L</i>	<i>P_M</i>	<i>P_H</i>
80	450	900	1800	690	1380	2760	1225	2450	4900
85	460	920	1840	800	1600	3200	1320	2640	5280
90	550	1100	2200	945	1890	3780	1420	2840	5680
95	570	1140	2280	1085	2170	4340	1520	3040	6080
100	580	1160	2320	1200	2400	4800	1640	3280	6560
105	650	1300	2600	1310	2620	5240	—	—	—
110	780	1560	3120	1420	2840	5680	—	—	—
120	790	1580	3160	1530	3060	6120	—	—	—
130	940	1880	3760	1590	3180	6360	—	—	—

<i>d</i> /mm	7000AC			7200AC			7300AC		
	<i>P_L</i>	<i>P_M</i>	<i>P_H</i>	<i>P_L</i>	<i>P_M</i>	<i>P_H</i>	<i>P_L</i>	<i>P_M</i>	<i>P_H</i>
10	40	80	160	75	150	300	110	220	440
12	40	80	160	90	180	360	130	260	520
15	45	90	180	105	210	420	145	290	580
17	55	110	220	140	280	560	180	360	720
20	80	160	320	175	350	700	220	440	880
25	90	180	360	200	400	800	330	660	1320
30	110	220	440	270	540	1080	400	800	1600
35	210	420	840	380	760	1520	525	1050	2100
40	220	440	880	435	870	1740	615	1230	2460
45	280	560	1120	480	960	1920	755	1510	3020
50	290	580	1160	500	1000	2000	880	1760	3520
55	405	810	1620	620	1240	2480	1070	2140	4280
60	430	860	1720	750	1500	3000	1240	2480	4960
65	440	880	1760	780	1560	3120	1400	2800	5600
70	530	1060	2120	850	1700	3400	1570	3140	6280
75	540	1080	2160	970	1940	3880	1710	3420	6840
80	665	1330	2660	1045	2090	4180	1850	3500	7400
85	685	1370	2740	1220	2440	4880	2010	4020	8040
90	850	1700	3400	1440	2880	5760	2150	4300	8600
95	875	1750	3500	1650	3300	6600	2300	4600	9200
100	895	1790	3580	1830	3660	7320	2620	5240	10480
105	1000	2000	4000	1995	3990	7980	—	—	—
110	1190	2380	4760	2160	4320	8640	—	—	—
120	1215	2430	4860	2330	4660	9320	—	—	—
130	1460	2920	5840	2415	4830	9660	—	—	—

(continued)

d/mm	7200B			7300B		
	P_L	P_M	P_H	P_L	P_M	P_H
20	175	350	700	—	—	—
25	195	390	780	320	640	1280
30	250	500	1000	400	800	1600
35	335	670	1340	470	940	1880
40	400	800	1600	580	1160	2320
45	445	890	1780	735	1470	2940
50	480	960	1920	840	1680	3360
55	570	1140	2280	970	1940	3880
60	690	1380	2760	1010	2020	4040
65	780	1560	3120	1270	2540	5080
70	865	1730	3460	1410	2820	5640
75	900	1800	3600	1620	3240	6480
80	990	1980	3960	1660	3320	6640
85	1150	2300	4600	1820	3640	7280
90	1310	2620	5240	1950	3900	7800
95	1485	2970	5940	2120	4240	8480
100	1600	3200	6400	2340	4680	9360
105	1765	3530	7060	2485	4970	9940
110	1895	3790	7580	2660	5320	10640

Note: 1. The light preload, the moderate preload and the heavy preload of the 71900C series bearings are 0.005, 0.015 and 0.030 times of the bearing dynamic rating load, respectively.

2. The bearing preloads for the inner ring diameter $d > 100\text{mm}$ that are not listed in Table 4-25 are as follows:

The light preload, the moderate preload and the heavy preload of the 7000C series bearings are 0.009, 0.018 and 0.036 times of the bearing dynamic rating load, respectively.

The light preload, the moderate preload and the heavy preload of the 7200C, 7300C series bearings are 0.01, 0.02 and 0.04 times of the bearing dynamic rating load, respectively.

The light preload, the moderate preload and the heavy preload of the 7000AC series bearings are 0.015, 0.030 and 0.060 times of the bearing dynamic rating load, respectively.

The light preload, the moderate preload and the heavy preload of the 7200AC, 7300AC, 7200B, 7300B series bearings are 0.016, 0.032 and 0.064 times of the bearing dynamic rating load, respectively.

3. P_L —light preload, P_M —moderate preload, P_H —heavy preload.

4.3.7 Radial Preload

The radial preload of rolling bearings is usually used to eliminate the initial radial floating clearance in bearing by using the interference fit to let the bearing inner ring expanding in order to get the preloading condition.

The main purpose of the radial preload is generally to increase the number of rolling elements under load and thus to reduce the maximum rolling element load.

It is also used to prevent the revolution sliding in high speed cylindrical roller bearings.

For the bearings of which the inner ring has tapered bore, if letting the inner ring moving along the axial direction to increase the interference fit amount between the inner ring and the shaft, the inner ring would be expanded, thus the bearing radial clearance would be reduced.

In this way the radial preload will be achieved.

4.4 Rolling Bearing Rotation Speed Limit

The rolling bearing rotation speed limit is the permissible maximum rotation speed under certain load, lubrication condition. That are concerned with many factors, such as the bearing type, the bearing dimensions, the load value and direction, the lubricant type and lubrication method, the bearing clearance, the cage structure and the cooling condition and so on.

It is very complicated to determine the rolling bearing rotation speed limit and there are no precise calculation methods for determining every type rolling bearing rotation speed limit. The only thing we can do is to provide some approximate calculation equation for the rolling bearing rotation speed limit according to the used experience and test results. Some opinions about reasonable usage also can be put forward. During selecting the rolling bearing, the rotation speed limit could not exceed the rolling bearing rotation speed limit value in the bearing catalog.

4.4.1 Determination Method of Rolling Bearing Rotation Speed Limit

The rolling bearing rotation speed limits given in the bearing catalog are determined under certain assuming conditions, that are as follows: assuming

that the dynamic equivalent load $P < 0.1C$ (C is the bearing dynamic rating load) ; normal lubrication, cooling conditions; for radial bearing, the subjected load is only radial load, for thrust bearing, the subjected load is only axial load; the bearing accuracy is the $P0$ grade.

Under the above assuming conditions, the rolling bearing rotation speed limit can be determined by the following equation:

For radial bearing

$$n_j = \frac{f_1 A}{D_m} \quad (4-75)$$

For thrust bearing

$$n_j = \frac{f_1 A}{\sqrt{DH}} \quad (4-76)$$

Where n_j — rolling bearing rotation speed limit (r/min) ;

D_m — average bearing diameter (mm) , $D_m = 0.5 (d + D)$;

d — bearing inner ring diameter (mm) ;

D — bearing outer ring diameter (mm) ;

H — thrust bearing height (mm) ;

f_1 — dimension factor, its value can be checked out from Figure 4-33 according to the radial bearing outer diameter D_m value or the thrust bearing $\sqrt{DH}$ value ;

A — structure factor, its value can be checked out from table 4-26.

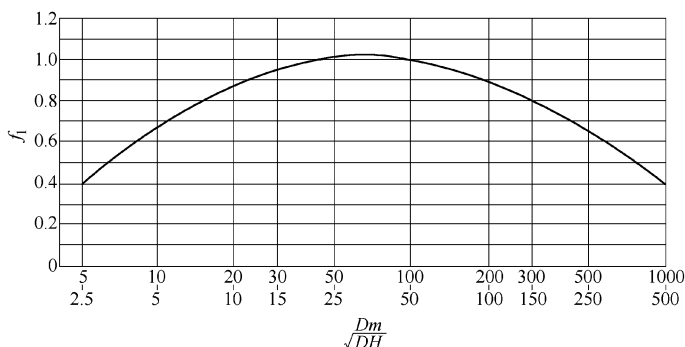


Figure 4-33 Dimension factor f_1

The calculation results should be rounded to two effective digits. When calculating the needle roller bearing rotation speed limit, the inner ring raceway diameter should be used instead of D_m in the equation (4-75). Table 4-27 lists the rolling bearing rotation speed limit value ranges.

Table 4-26 Structure factor A

Bearing type	Grease lubrication	Oil lubrication
deep groove ball bearing		
single row	48	60
single row with shield	48	60
single row with seals	34	—
single row with felt seal	24	—
single row with filling slot	38	48
double row with filling slot	30	38
angular contact ball bearing		
single row	45	60
double row	32	43
paired mounting	32	43
four point contact	36	48
separable type		
(magneto bearing)	48	60
cylindrical roller bearing	43	53
needle roller bearing		
without cage	9	12
with cage	24	36
drawn cup with cage	20	28
self-aligning roller bearing	28	34
tapered roller bearing		
single row	22	28
double row	22	28
four row	18	22
thrust ball bearing	9	13
thrust cylindrical roller bearing	6, 7	9
thrust tapered roller bearing	6, 7	9
thrust self-aligning roller bearing	—	18

Table 4 – 27 Rolling bearing rotation speed limit value range

10	38	100	380	1000	3800	10000	38000
11	40	110	400	1100	4000	11000	40000
12	43	120	430	1200	4300	12000	43000
13	45	130	450	1300	4500	13000	45000
14	48	140	480	1400	4800	14000	48000
15	50	150	500	1500	5000	15000	50000
16	53	160	530	1600	5300	16000	53000
17	56	170	560	1700	5600	17000	56000
18	60	180	600	1800	6000	18000	60000
19	63	190	630	1900	6300	19000	63000
20	67	200	670	2000	6700	20000	67000
22	70	220	700	2200	7000	22000	70000
24	75	240	750	2400	7500	24000	75000
26	80	260	800	2600	8000	26000	80000
28	85	280	850	2800	8500	28000	85000
30	90	300	900	3000	9000	30000	90000
32	95	320	950	3200	9500	32000	95000
34	100	340	1000	3400	10000	34000	100000
36		360		3600		36000	

4.4.2 Rolling Bearing Rotation Speed Limit Actually Used

There are many factors for influencing the rolling bearing rotation speed limit. The mainly influence factors are as follows:

(1) Load value

While the bearing runs under the condition of the dynamic equivalent load $P > 0.1C$, the contact stress on the contact area of the rolling element and the raceway will be increased, the temperature will be risen that will influence the lubricant performance.

Therefore, the load factor f_2 should be introduced in the rolling bearing rotation speed limit given in the bearing catalog as shown in Figure 4 – 34.

(2) Load type and direction

If both the radial load and the axial load are applied to the radial bearing at the same time, because the number the loading rolling elements

will be increased, and the friction heat will also be increased, the load distribution factor f_3 should multiply the rolling bearing rotation speed limit given in the bearing catalog as shown in Figure 4-35.

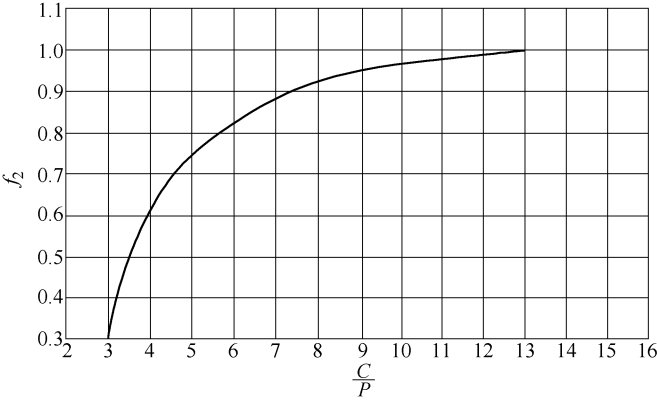


Figure 4-34 Load factor f_2

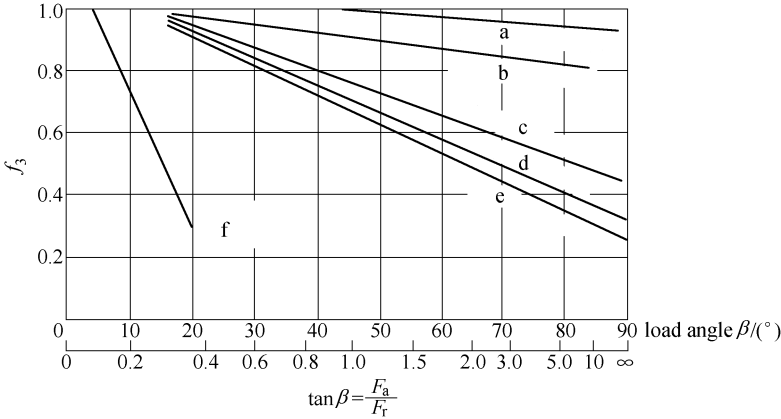


Figure 4-35 Load distribution factor f_3

- a — Angular contact ball bearing b — Single radial ball bearing
c — Tapered roller bearing d — Aligning ball bearing
e — Self-aligning roller bearing f — Cylindrical roller bearing

(3) Lubricant and lubrication method

The rolling bearing rotation speed limits given in the bearing catalog are

suitable for ordinary lubrication condition, such as the oil bath or the dropping lubrication. If using the circulating oil lubrication, oil mist lubrication, spraying lubrication or oil gas lubrication, the bearing rotation speed limit could be raised to 1.5 ~ 2 times.

The used experience has proved that the bearing rotation speed limit could also be raised by increasing the bearing manufacturing accuracy and the bearing clearance, using the cage made by the special material and special structure.

4.4.3 Test Method of Rolling Bearing Rotation Speed Limit

Because of the complication of the bearing rotation speed limit, during selecting the bearing rotation speed limit for the main machine bearings, the bearing rotation speed limit test should be carried out.

(1) Test condition

1) Test machine The bearing rotation speed limit test must be performed on the tester recognized by China bearing industry standard.

2) Sample Every batch testing bearings must be the same batch manufacturing and the same model bearings. The 15 ~ 20 test bearings should be prepared and the 10 bearings within them are tested using the same test regulation and on the same structure testers.

3) Test bearing load The test bearing load should be as:

$$P = 0.1C$$

Where P — bearing equivalent dynamic load (N);

C — bearing dynamic rating load (N).

During the testing, the test load error should be controlled in the range of $\pm 2\%$.

4) Testing rotation speed The testing of bearing of rotation speed should be selected according to rolling bearing rotation speed limit given in the bearing catalog. The test rotation speed error should be controlled in the range of $\pm 2\%$.

5) Test bearing lubrication The test bearing lubrication grease and the lubrication oil must be conformed with the National Standard.

The oil impurity should not be larger than $10\mu\text{m}$. The oil supplying

amount should be not less than $1.7 \sim 2.5 \text{ L/s}$. The grease filling amount should be $1/4 \sim 1/2$ of the space volume, even filling.

(2) Test methods

1) Fixed time inspection test Under the same test condition, the sample test will be stopped after testing 24 hours.

2) Failure judgement If the sample testing of bearing outer ring temperature is over 100°C (the temperature rise is over 60°C), then the sample is failed.

3) Others During the test, if there appear the following phenomena, such as leaning to the rings, wear, breaking, rivet loosening or breaking and the obvious increase of the bearing clearance, serious lowering of running accuracy, the obvious increase of bearing vibration and noise, or there appear color changing, burning, blocking, then the test should be stopped at once.

(3) Test result analysis

The passing — through rate of the rotation speed limit through the testing is:

$$R_a = \frac{N - N_1}{N} * 100\%$$

Where N — testing bearing numbers;

N_1 — the bearing numbers less than 24 hour testing;

R_a — the passing — through rate of the rotation speed limit through the testing.

4.5 Minimum Axial Load of Thrust and Thrust Angular Contact Bearing

A body that its implicative moving is the circumference moving, while its self-rotating axis direction is always changing, there is an inertia moment acting on this body, or calls as gyroscopic moment.

The contact angles of both the angular contact ball bearing and the roller bearing are not equal to zero. Under their running conditions, the self-rotating axis direction of the rolling element is always changing.

Therefore, under the stably running condition, the rolling element center are subjected to the centrifugal force and the inertia moment.

The direction and value are as:

$$M_K = J_0 \omega_w \omega_c \sin \alpha \quad (4-77)$$

Where M_K — inertia moment ($\text{N} \cdot \text{m}$)

J_0 — rolling element rotation inertia value;

ω_w — rolling element self-rotating angle speed;

ω_c — rolling element revolution angle speed

α — bearing contact angle.

For ball bearing

$$J_0 = \frac{1}{60} \frac{\pi \gamma}{g} D_w^5 \quad (4-78)$$

Where γ — ball material specific gravity;

g — acceleration of gravity;

D_w — ball diameter.

If the ball is made by the steel:

$$J_0 = 4.2 * 10^{-8} D_w^5$$

Figure 4-36 shows the inertia moment direction on the rolling element in thrust angular contact roller bearing. Figure 4-37 shows the inertia moment direction on the rolling element in thrust ball bearing. Figure 4-38 shows the inertia moment direction on the rolling element in angular contact ball bearing.

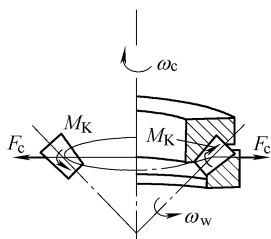


Figure 4-36 Inertia moment direction in thrust angular contact roller bearing

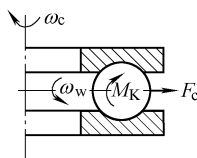


Figure 4-37 Inertia moment direction in thrust ball bearing

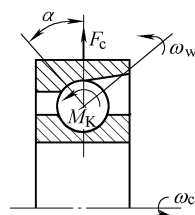


Figure 4-38 Inertia moment direction in angular contact ball bearing

It is known from the equation (4-77) that the inertia moment is proportional to the contact angle sine. It is also known from the equation (2-143) and equation (2-146) that ω_w and ω_c are in proportional to the bearing rotation speed n .

Therefore, if the contact angle is much larger and the bearing rotation speed is much higher, then the inertia moment is much larger. The inertia moments of thrust bearing and thrust radial bearing are the largest. When the inertia moment M_k is larger than the friction moment between the raceway and the rolling element, the self-rotating sliding would be happened.

In thrust ball bearing, if the shaft washer is running in rotation speed n and the housing washer is in the stationary, then from the equation (2-146), the following equation can be got:

$$\omega_w = \frac{2\pi}{60} n_w = \frac{\pi}{60} \frac{D_m}{D_w} n$$

From the equation (2-143), the following equation can be got:

$$\omega_c = \frac{2\pi}{60} n_c = \frac{\pi}{60} n$$

If substituting J_0 , ω_w and ω_c into the equation (4-77), the following equation can be got:

$$\begin{aligned} M_k &= 1.15 * 10^{-10} D_m D_w n^2 \\ &= 1.15 * 10^{-16} D_m D_w n^2 \\ &= 11.3 * 10^{-16} D_m D_w n^2 \end{aligned}$$

We can prevent the self-rotating sliding caused by the inertia moment by increasing the friction moment between the raceway and the rolling element. The condition to avoid the self-rotating sliding is:

$$M_f > M_k$$

M_f is the friction moment between the raceway and the rolling element caused by the axial load. For every kind of thrust and thrust radial bearings, the minimum axial load calculation equations are listed in Table 4-28.

Table 4 – 28 Minimum axial load F_{amin}

Bearing type	F_{amin}
thrust ball bearing	$F_{amin} > A \left(\frac{n}{1000} \right)^2$
thrust angular contact ball bearing $\alpha = 45^\circ$	$F_{amin} > 0.19F_r + A \left(\frac{n}{1000} \right)^2$
$\alpha = 65^\circ$	$F_{amin} > 0.33F_r + A \left(\frac{n}{1000} \right)^2$
thrust self-aligning roller bearing	$\frac{C_0}{10^4} \leq F_{amin} > 0.18F_r + A \left(\frac{n}{1000} \right)^2$
thrust roller bearing	$\frac{C_0}{10^4} \leq F_{amin} > A \left(\frac{n}{1000} \right)^2$

In Table F_{amin} — minimum axial load;

F_r — radial load (N);

α — contact angle;

n — bearing rotation speed (r/min);

C_0 — static rating load (N);

A — minimum axial load constant calculated from the equation (4-79),

$$A = k \left(\frac{C_0}{10^4} \right)^2 \quad (4-79)$$

The factor k value can be checked out from Table 4 – 29.

Table 4 – 29 Factor k value

Bearing type	k
thrust ball bearing	0.01
thrust angular contact ball bearing	0.004
thrust self-aligning roller bearing	0.0004
thrust roller bearing	0.0004
thrust tapered roller bearing	0.0002

For most applications, the actual axial load of thrust bearing is larger than the minimum axial load calculation value, thus the bearing is not required to preload.

While the actual axial load of thrust bearing is smaller than the minimum axial load calculation value, the bearing must be preloaded.

For thrust ball bearings, on some conditions, it is inevitably sliding between the balls and the raceways due to inertia moment. For example, in double-row angular contact ball bearings, one row of balls are generally not subjected to the load. Therefore, it is inevitably sliding between the balls and the raceways in the row that is not subjected to the load. The used experience has proven that if the actual axial load of thrust bearing is smaller than $0.000016C_0$, sliding between the balls and the raceways could not cause the bearing failure. In this case, the bearing is not required to preload.

4.6 The Axial Load Capacity of Cylindrical Roller Bearings

The NJ type, Nup type and NF type of the cylindrical roller bearings with ribs both in the inner rings and in the outer rings, are mainly used not only in carrying radial load, but also in carrying certain single direction axial load. Therefore, these bearings can be used as fixed support.

The axial load carrying capacity of these bearings is decided by the load carrying capacity of the sliding friction faces between the roller end faces and the ring guiding ribs, that is decided by the lubrication condition, the working temperature, and the bearing heat dissipation condition.

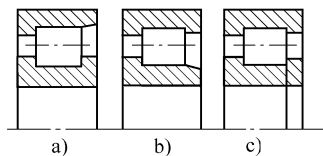


Figure 4-39 Cylindrical roller bearings carrying axial load

a) NF type b) NJ type c) NUP type

The following experimental equation can be used as the guide for evaluating the axial load carrying capacity while selecting these bearings:

$$F_{ap} = \frac{K_1 C_0 * 10^4}{n(d + D)} - K_2 F_r \quad (4-80)$$

Where F_{ap} — maximum permissible axial load (N) ;

C_0 — basic static rating load (N) ;

F_r — bearing radial load (N) ;

- n — bearing rotation speed (r/min);
 d — bearing bore diameter (mm);
 D — bearing outer ring diameter (mm);
 K_1, K_2 — factors shown in Table 4-30.

Table 4-30 Factors K_1, K_2

Bearing type	Grease lubrication		Oil lubrication	
	K_1	K_2	K_1	K_2
EC type	1.5	0.15	1	0.1
other bearings	0.5	0.05	0.3	0.03

The above equation is suitable for the following bearing working conditions:

- 1) The axial load on the bearing is a continuously acted stable load;
- 2) The bearing working temperature difference with the surrounding temperature is 60°C ;
- 3) The power loss caused by the bearing temperature rise is $0.5 \text{ mW/mm}^2 \cdot ^{\circ}\text{C}$.
- 4) The viscosity ratio is $k=2$.

The viscosity ratio is the rate of the lubricant viscosity ν under the bearing working temperature with that of the required lubricant viscosity ν_1 under the bearing working temperature for getting the good lubrication.

If the axial load acts just for a short time, the calculated value should be multiplied by 2; if the axial load is a shocking load, the calculated value should be multiplied by 3.

The stable axial load F_a acting on bearings can not exceed $1.2D^2\text{N}$ (D — bearing outer diameter, unit: mm, here is just using its value). If there is just occasionally shocking load, its value can not exceed $3D^2\text{N}$.

In order to guarantee the bearing rib uniform loading and the spindle running accuracy, the high requirement for the axial runout of the neighboring component surface contacting with the bearing also should be put forward.

The axial loads on two roller ends form a moment because of their acting lines are not coincided with the roller central line, as shown in Figure

4 – 40. This moment will make the roller tilting in the bearing axial plane, thus, the load distribution along the roller length will be changed. The larger the $\frac{F_a}{F_r}$ is, the more serious the tilt is.

Therefore, the short cylindrical roller bearings can not carry pure axial load, that is when subjecting to axial load, the bearings must be subjecting to radial load at the sametime.

In this case, the bearing can normally work. The experience has been proven that while $F_a < 0.4 F_r$, the bearing can be kept normally working.

Theoretically saying that, the short cylindrical roller bearing life should be influenced by the axial load in certain level. But while $\frac{F_a}{F_r} < 0.13$, the axial load just has a little influence on bearing life.

Generally, it is not considered.

Besides, under the axial load, the friction forces between the ribs of the inner ring and the outer ring and the roller end faces also form a moment that would be tending to let the roller rotating in the tangent plane of roller axis. This moment increases with the roller length. Therefore, the long cylindrical roller bearings and the needle roller bearings are not suitable for carrying the axial load.

The cylindrical roller bearing axial load capacity is directly influenced by the axial load characteristics. Usually, the interrupted axial load could improve the lubrication condition.

Therefore, the cylindrical roller bearing axial load capacity under the interrupted axial load could be raised about 1 ~ 2 times of the calculated value by the above equation.

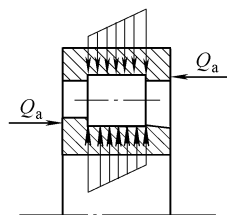


Figure 4 – 40 Axial load forming moment about rolley

4. 7 Required Minimum Radial Load F_{rmin} for Radial Rolling Bearing

During radial bearing running, because of the influence of the inertia

forces of the cage and the rolling elements and the lubricant, the damaging sliding movements between the rolling elements and the raceways may be occurred.

Particularly in high speed rotating bearings, this sliding friction may cause bearing early damage.

Usually this phenomenon is called as light load sliding. In order to prevent the light load sliding, the radial bearings must be subjected to a given minimum radial load $F_{\min}$.

4.7.1 Required Minimum Radial Load $F_{\min}$ for Deep Groove ball Bearings, Self-Aligning Ball Bearings and Angular Contact Ball Bearings

The required minimum radial load for these type bearings can be evaluated by the following equation:

$$F_{\min} = K_r \left(\frac{\nu n}{1000} \right)^{2/3} \left(\frac{D_m}{100} \right)^2$$

Where $F_{\min}$ — required minimum radial load (N);

K_r — minimum load factor given in Table 4 – 31;

ν — oil viscosity at bearing operating temperature (mm^2/s);

n — bearing rotational speed (r/min);

D_m — bearing average diameter (mm).

Table 4 – 31 Minimum load factor K_r

Bearing type	K_r
deep groove ball bearing	618 series, taking $K_r = 15$
	619, 160 series, taking $K_r = 20$
	60, 161, 62 series, taking $K_r = 25$
	63 series, taking $K_r = 30$
	64 series, taking $K_r = 35$
double row deep groove ball bearing	42 series, taking $K_r = 50$
	43 series, taking $K_r = 60$
paired mounting deep groove ball bearing	taking the double K_r value
filling slot full complement deep groove ball bearing	200 series, taking $K_r = 40$
	300 series, taking $K_r = 50$

(continued)

Bearing type	K_r
self aligning ball bearing	14 , 12E, 13E, 22 series , taking $K_r = 30$ 113 , 112 series , taking $K_r = 40$ 22E, 13 series , taking $K_r = 45$ 23E series , taking $K_r = 50$
double-row angular contact ball bearing	32 series , taking $K_r = 80$ 32A series , taking $K_r = 60$ 32E series , taking $K_r = 90$ 33 series , taking $K_r = 95$ 33A series , taking $K_r = 70$ 33E series , taking $K_r = 110$
DB or DF arrangement pair matched pair angular contact ball bearing	72B series , taking $K_r = 80$ 72BE series , taking $K_r = 95$ 73B series , taking $K_r = 90$ 73BE series , taking $K_r = 100$

4. 7. 2 Required Minimum Radial Load F_{rmin} for Cylindrical Roller Bearings

The required minimum radial load for the cylindrical roller bearings can be evaluated by the following equation:

$$F_{\text{rmin}} = K_r \left(6 + \frac{4n}{n_j} \right) \left(\frac{D_m}{100} \right)^2$$

- where F_{rmin} — required minimum radial load (N) ;
 K_r — minimum load factor given in Table 4 – 32 ;
 n — bearing rotational speed (r/min) ;
 n_j — bearing rotational speed limit in oil lubrication (r/min) ;
 D_m — bearing average diameter (mm) .

Table 4 – 32 Minimum load factor K_r

Bearing type	K_r
radial cylindrical roller bearing	10 series, taking $K_r = 100$
	2, 3, 4 series, taking $K_r = 150$
	22 series, taking $K_r = 200$
	23 series, taking $K_r = 250$
full complement radial cylindrical roller bearing	18 series, taking $K_r = 100$
	29, 48 series, taking $K_r = 200$
	49 series, taking $K_r = 250$
	22, 30 series, taking $K_r = 300$
	23 series, taking $K_r = 350$
	NNF50 series, taking $K_r = 400$
	50 series, taking $K_r = 500$

4.7.3 Required Minimum Radial Load for Needle Roller Bearings, Self-Aligning Roller Bearings and Tapered Roller Bearings

The required minimum radial load for needle roller bearings, self-aligning roller bearings, tapered roller bearings can be evaluated by the following equation:

$$F_{\min} = 0.02C$$

4.7.4 Required Minimum Radial Load for Single Angular Contact Ball Bearings, DT Arrangement Angular Contact Ball Bearings and Four-Point Contact Ball Bearings

The required minimum radial load for a single angular contact ball bearing, DT arrangement angular contact ball bearing, four point contact ball bearing can be evaluated by the following equation:

$$F_{\min} = K_a \left(\frac{C_0}{100} \right) \left(\frac{nD_m}{100000} \right)^2$$

where $F_{\min}$ — required minimum radial load (N);

K_r — minimum load factor given in Table 4 – 33;

n — bearing rotational speed (r/min);

D_m — bearing average diameter (mm).

In actual bearing usage, if the load applied on the bearing is smaller than the calculated load value evaluated by the above equation, in this case, for preventing light sliding, the bearing must be preloded. The additional axial load can be got by spring preload, and the additional radial load can be got by tensing the transmission belt.

Table 4 –33 Minimum axial load factor K_a

Bearing type	K_a
Single angular contact ball bearing, DT arrangement angular contact ball bearing	72BE series, taking $K_a = 1.4$
	72B series, taking $K_a = 1.2$
	73BE series, taking $K_a = 1.2$
	73B series, taking $K_a = 1.4$
four point contact ball bearing	QJ2 series, taking $K_a = 1.0$
	QJ3 series, taking $K_a = 1.1$

Chapter 5 Rolling Bearing Vibration and Noise

Rolling bearing vibration and noise levels are the important bearing dynamic characteristics, comprehensively and fully reflecting the rolling bearing product quality level and mostly concerned by the rolling bearing users. Therefore, this chapter mainly describes the basic principle and the China existing measuring methods, the China standards and the ISO standards about rolling bearing vibration and noise.

The symbols used in this chapter are as follows :

f_r — inner ring rotation frequency (Hz) ;

f_i — rolling element set (or cage) rotation frequency relative to the inner ring (Hz) ;

$$f_i = \frac{1}{2}f_r \left(1 + \frac{D_b}{D_m} \cos \alpha \right)$$

f_e — rolling element set (or cage) rotation frequency relative to the outer ring (Hz) ;

$$f_e = \frac{1}{2}f_r \left(1 - \frac{D_b}{D_m} \cos \alpha \right)$$

f_b — rolling element rotation frequency in the cage system (Hz).

$$f_b = \frac{f_r D_m}{2D_b} \left(1 - \frac{D_b^2}{D_m^2} \cos^2 \alpha \right)$$

In above equations: D_b — rolling element diameter;

D_m — bearing average diameter.

Rolling Bearing vibration and noise levels comprehensively and fully reflect the rolling bearing product dynamic quality, thus, many nations have given special attention to these issues and have stipulated control standards and measuring standards.

During rolling bearing rotational running, all normal movements except the movements that deviate from theoretical position are called as rolling bearing vibration. The radiation wave of bearing vibration in air medium is called as noise.

Table 5 - 1 Elastic inherent vibration type of rings

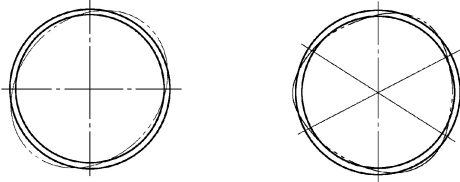
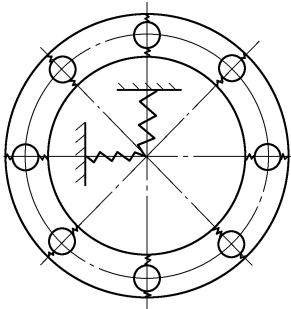
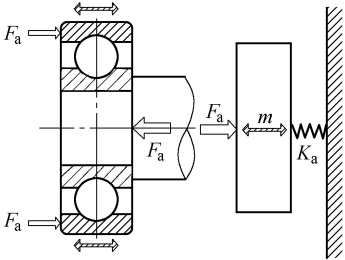
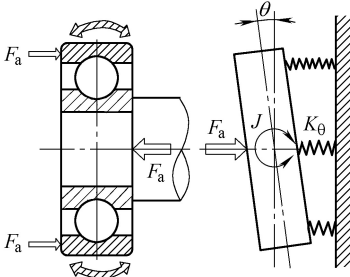
Vibration type	Ring radial elastic flexibility vibration	ring radial elastic flexibility vibration
sketch	 first second	 first second
inherent frequency	$f_n = \frac{1}{2\pi} \frac{(n^2 - 1) \frac{\pi EI}{R^2} + \frac{K_N z}{2}}{\pi \rho A R \left(1 + \frac{1}{n^2} \right)}$ $n \leq \frac{z - 1}{2}$	—
note	assumed that the outer ring is a round ring restrained by springs	this type vibration amplitude is very small
symbols	f_n —ring n step inherent frequency K_N —elastic factor R —ring average radius (mm) A —ring section area (mm ²) I —ring section second moment (mm ⁴)	E —elasticity modulus (N/mm ²) ρ —density (g/mm ³) z —ball number n —integral

Table 5-2 Ring rigidity inherent frequency

Vibration type	Ring radial rigidity vibration	Ring axial rigidity vibration	Ring angle rigidity vibration
sketch			
inherent frequency /Hz	$f = \frac{1}{2\pi} \left(\sqrt{\frac{zK_0}{2m}} \right)$	$f = \frac{1}{2\pi} \left(\sqrt{\frac{K_a}{m}} \right)$	$f = \frac{1}{2\pi} \left(\sqrt{\frac{K_\theta}{J}} \right)$
note	assumed that all rolling elements are elastic contact with the ring		
symbols	m—ring mass (kg) K_0—equal effective contact rigidity (N/m) of the contact point of rolling element with inner ring and outer ring K_a—ring axial direction contact rigidity (N/m) K_θ—ring angle contact rigidity (N/rad) J—ring section plane moment of ring section plane for diameter		

5.1 Rolling Bearing Vibration Characteristics

5.1.1 Rolling Bearing Vibration Types

According to the bearing vibration direction, there are three types:

- 1) Radial direction vibration The vibration in the bearing ring radial plane.
- 2) Axial direction vibration The vibration in the bearing ring axial plane.
- 3) Angle direction vibration The vibration in the bearing ring angle plane.

According to the bearing vibration mechanism, rolling vibration can be divided as three types:

(1) Bearing structure inherent vibration

- 1) The inherent vibration caused by the outer ring considered as an elastic body, Table 5 - 1.
- 2) The inherent vibration caused by the outer ring considered as a rigidity body, Table 5 - 2.
- 3) The inherent vibration caused by the ball considered as a rigidity body, Table 5 - 3.

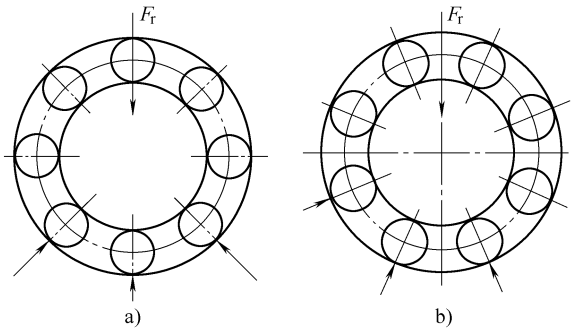
Table 5 - 3 Rolling element inherent vibration

Vibration type	Rolling element inherent vibration	Symbolic meaning
inherent frequency/Hz	$f = \frac{0.848}{D_b} \sqrt{\frac{E}{2\rho}}$	D_b — ball diameter (mm) E — elasticity modulus (N/mm ²) ρ — density (g/mm ³)

(2) Forced vibration

- 1) Rolling element through vibration When the bearing is rotational running, the rolling elements periodically pass through the force acting line, causing the rolling element through vibration, Table 5 - 4.

Table 5-4 Rolling element through vibration

Vibration type	Rolling element through vibration
sketch	 a) b)
vibration frequency /Hz	$f = z f_c$
Note	① As shown in the sketch, during the rolling element rotational running, the rolling element load condition changes with its position, so, the alternate changing elastic force is produced. Therefore, the periodical vibration is produced when it passes through the radial load acting line. ② This type vibration amplitude increases with the radial load and the bearing floating clearance. While the bearing is only subjected to the axial load, this type vibration amplitude is small.

2) Bearing vibration caused by bearing component manufacturing error, Table 5-5.

Table 5-5 Bearing vibration caused by bearing component manufacturing error^①

Manufacturing error	Waviness number	Radial direction (angle) vibration frequency/Hz	Axial direction vibration frequency/Hz
inner ring waviness	$K = nz$ $K = nz \pm 1$	— $Kf_i \pm f_c$	Kf_i —
outer ring waviness	$K = nz$ $K = nz \pm 1$	— $(K \pm 1) f_c$	Kf_c —

(continued)

Manufacturing error	Waviness number	Radial direction (angle) vibration frequency/Hz	Axial direction vibration frequency/Hz
ball waviness	$K = nz$	$Kf_b \pm f_c$	Kf_b
ball diameter difference		f_c	

① Bearing outer ring vibration while the inner ring is running and the outer ring is stationary.

(3) Shocking vibration

The pulsing vibration of bearings is stimulated by the defects of score , burr , rust , pitting , flaking and etc. on the raceway surface or on the rolling element surface. The pulse period is in reverse proportional to the rotation speed and the pulse amplitude is related to the defect dimensions. This type vibration pulse number per second is shown in Table 5 – 6.

Table 5 – 6 Raceway surface defects stimulated vibration frequency

Factors stimulating bearing vibration	Vibration frequency/Hz	Vibration feature
inner ring raceway one pit defect	zf_i	forming equal time interval pulse vibration
outer ring raceway one pit defect	zf_c	forming equal time interval pulse vibration
rolling element surface one pit defect	zf_b	forming equal time interval pulse vibration

The rolling bearing vibration always is the jointly acting result of the above vibration types. A large amount of research data have proven that under normal bearing running condition , the bearing component waviness is the main factor causing the bearing vibration.

5. 1. 2 Factors Influencing Rolling Bearing Vibration

1. Design parameter influence on rolling bearing vibration

The rolling bearing structure is mainly determined according to the load

carrying capacity. But besides meeting other characteristics, the different bearing structure vibration feature is different. The most influential factors are rolling element number, ring wall thickness and clearance, as shown in Table 5 – 7.

Table 5 – 7 Design parameter influence on rolling bearing vibration

Design parameter	Design parameter influence on rolling bearing vibration
rolling element number	① The ring and rolling element waviness that stimulates rigidity vibration amplitude of the ring is in proportional to $\frac{1}{\sqrt{z}}$ ② The rolling element load that stimulates flexibility vibration amplitude is in proportional to $\frac{1}{z^2}$ ③ The ring inherent frequency vibration amplitude is in proportional to $\sqrt{z}$
ring wall thickness	① The ring wall thickness is in proportional to the flexibility rigidity. Therefore, to increase the ring wall thickness can reduce the ring flexibility vibration. ② The ring rigidity inherent frequency is in reverse proportional to the square root of the ring mass $\frac{1}{\sqrt{m}}$. Therefore, to increase the ring wall thickness can reduce the ring vibration frequency.
clearance	① The rolling element through vibration increases with the radial clearance ② During bearing running, there is an optimum clearance, at which the bearing vibration and noise level is the smallest, but the value must be determined by test.

2. Bearing component manufacturing errors influence on bearing vibration

The main bearing component manufacturing errors are roughness, waviness and roundness. A large amount of experiments have proved that the bearing component waviness is the main factor causing the bearing vibration, but the influence of the roughness and roundness are little. The bearing elastic flexibility vibration and the rigidity vibration are both reduced by the improvement of the waviness error.

3. Working condition influence on bearing vibration

During bearing running, the largest influential factors for bearing vibrations are load, rotation speed and lubrication condition, as shown in Table 5 – 8.

Table 5 – 8 Working condition influence on bearing vibration

Working condition	Influence on bearing vibration
load	① The inherent vibration frequency increases with the load. ② The vibration amplitude is in proportional to the load 6/5th power. ③ Proper preload is needed for the low noise bearing, in order to guarantee that all of the rolling elements can contact with the raceway. The high frequency vibration would be caused if the ball is loosen in the non-loading area.
rotation speed	① In the vibration frequency spectrum, the rotation speed an the forced vibration frequency and amplitude are in proportion to the rotation speed. ② The inherent frequency is not influenced by the rotation speed.
lubrication	① Lubrication can not eliminate certain frequency vibration, but the oil film formed by lubricant can alleviate the shock action between the rolling element and the ring raceway and can reduce the vibration amplitude. ② With the oil, grease viscosity increasing, the vibration amplitude will be lowered, mainly in the middle and high frequency range.

4. Installation factor influence on bearing vibration

The rolling bearing installes in the housing, its vibration model can be simplified as shown in Figure 5 – 1.

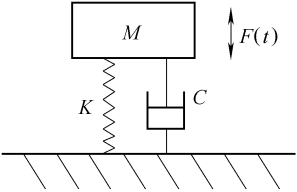


Figure 5 – 1 Bearing and housing simplified dynamics model

K , C — bearing and housing elastic contact rigidity and damping coefficient
 M — bearing system mass $F(t)$ — bearing stimulated vibration force

Table 5 – 9 shows the installation factor influence on bearing vibration.

Table 5 – 9 Installation factor influence on bearing vibration

Installation factor	Influence on bearing vibration
contact rigidity and damping	① Through proper selection of the bearing housing rigidity, the axial direction vibration amplitude and the radial direction amplitude of any special frequency can be lowered to the minimum value ② The damping component made by laminated structure can be successfully used for diminishing high frequency vibration
the fit between bearing outer ring and housing the fit between bearing inner ring and shaft fit	Through the change of the bearing working clearance influences on the bearing vibration. Varied bearing system optimum working clearance should be determined through the test.
eccentric, noncentral	① Producing larger rolling element load, stimulating the rolling element through vibration ② Stimulating multiple frequency rotation speed vibration

5.2 Rolling Bearing Vibration Measurement

Rolling bearing vibration is a very complicated physical phenomenon. The measurement data under certain test condition can not judge the bearing vibration characteristics of different working condition. But it is effective to select the bearing according to the present comprehensively adopted bearing vibration measurement method and standard of the world or China, which effectiveness has been proved by a large amount of tests.

At present, China accepted two physical quantities to express the bearing vibration levels: the velocity and the acceleration, and correspondingly developed two type bearing vibration testers: the velocity type bearing vibration tester and the acceleration type bearing vibration tester, and also have made two corresponding standards: the velocity type bearing vibration standard and the acceleration type bearing vibration standard. In the world, most of countries adopt the vibration velocity, but some countries adopt the

acceleration. Recently ISO published the International Standards: ISO15242 — 1: 2004 (The rolling bearing vibration measurement method — first part: basis) and ISO15242 — 2: 2004 (The rolling bearing vibration measurement method — second part: cylindrical bore and outside surface of radial ball bearing), its measuring physical quantity is vibration velocity $v_{r.m.s}$, the unit is $\mu\text{m/s}$, the measuring range is $50 \sim 10000\text{Hz}$, the frequency range are $50 \sim 300\text{Hz}$, $300 \sim 1800\text{Hz}$ and $1800 \sim 10000\text{Hz}$. Details are in Section 5.4 of this chapter.

5.2.1 Evaluating the Physical Quantities and Parameters of Rolling Bearing Vibration

In bearing vibration measurement, the commonly used physical quantities are listed in Table 5 – 10.

Table 5 – 10 Commonly used bearing vibration physical quantities

Physical quantity	Velocity	Anderon ^①	Acceleration
expression	$v = \frac{dR(t)}{dt}$	$A = \frac{dR(t)}{\sqrt{N}d\theta}$	$a = \frac{d^2R(t)}{dt^2}$
meaning	bearing vibration displacement change amount per unit time in one direction	bearing vibration displacement change amount per ring rotation unit angle	differential for bearing vibration velocity
unit	$\mu\text{m/s}$	$\mu\text{in/rad}$	$\mu\text{m/s}^2$ or $\text{dB}^{\text{②}}$

① For the vibration measurement system of which the rotation speed is as 1800r/min and multiple frequency factor $N = 2.5$, $1A = 7.5\mu\text{m/s}$.

② $L = 20 \lg (\frac{a}{a_0})$

In the equation: L — conversion of vibration value (dB);
 a — measured bearing vibration acceleration;
 a_0 — reference acceleration value. In the JB /T 5314 — 2002, the a_0 value is stipulated as $9.81 * 10^{-3} \text{ m/s}^2$.

After determining one physical quantity as measuring bearing vibration unit, the parameters listed in Table 5 – 11 can be used to evaluate the bearing vibration level.

Table 5 – 11 Evaluating the bearing vibration level parameters

Parameter	Mean value	Maximum peak value	Average square root value	Peak value factor
expression	$\mu = \frac{1}{T} \int_0^T R(t) dt$	$\max R(t)$	$\text{RMS} = \sqrt{\frac{1}{T} \int_0^T R^2(t) dt}$	$\frac{A_{\max}^{①}}{\text{RMS}}$

① $A_{\max}$ — The maximum peak value in the vibration frequency spectrum.

5. 2. 2 Devices Measuring Bearing Vibration

There are four parts for all of the measuring bearing vibration devices regardless of adopting any physical quantity to express the rolling bearing vibration: transducer system, measuring amplifier, driving spindle and loader. Figure 5 – 2 shows the velocity type vibration measuring device working principle.

In order to guarantee the bearing vibration measuring accuracy, china standards stipulated related technical requirements for any parts in the measuring system.

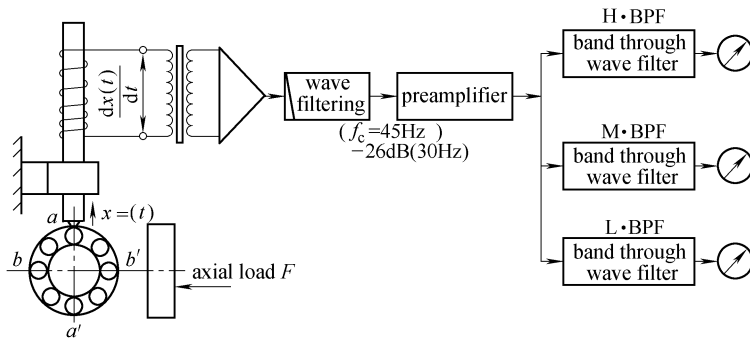


Figure 5 – 2 Rolling bearing vibration measuring principle

1. Measuring system base vibration

We starting the driving spindle (every measuring range switch is on the minimum place), pressing the transducer head down to let it on the same condition as the measuring condition. The velocity type vibration tester base vibration should conforms to the value in Table 5 – 12, and the acceleration type vibration tester base vibration should conform to the value in Table 5 – 13.

Table 5 – 12 Velocity type vibration tester base vibration (unit: $\mu\text{m/s}$)

Bearing nominal bore diameter/mm		Every frequency band vibration value max		
over	to	50 ~ 300Hz	300 ~ 1800Hz	1800 ~ 10000Hz
3 ^①	12	10	7	4
12	60	12	10	5
60	120	15	15	7

① Include 3mm.

Table 5 – 13 Acceleration type vibration tester base vibration

Bearing nominal bore diameter/mm	Base vibration/dB
$3 \leq d \leq 60$	< 10
$60 < d \leq 120$	< 15

2. rotation speed

During the bearing measuring, the velocity type vibration tester inner ring actual rotation speed n should be conformed to the value in Table 5 – 14, and the acceleration type vibration tester inner ring actual rotation speed n should be conformed to the value in Table 5 – 15.

Table 5 – 14 Velocity type vibration tester inner ring actual rotation speed

Bearing nominal bore diameter/mm		Inner ring actual rotation speed $n/(\text{r/min})$
over	to	
3 ^①	60	1764 ~ 1818
60	120	882 ~ 909

① Include 3mm .

**Table 5 – 15 Acceleration type vibration tester inner
ring actual rotation speed**

Bearing nominal bore diameter/mm	Spindle actual rotation speed
$3 \leq d \leq 60$	$1500\text{r/min} \pm 30\text{r/min}$
$60 < d < 120$	$1000\text{r/min} \pm 20\text{r/min}$

3. Mandrel

For velocity type vibration tester, after combining the mandrel with the spindle, the radial runout where the mandrel is fitted with the inner ring is

not larger than $5\mu\text{m}$, the axial runout of the mandrel with respect to the shaft abutment end face is not larger than $10\mu\text{m}$, the mandrel hardness should be 61 ~ 64 HRC. The fitting tolerance of the mandrel with the bearing bore should be conformed with the values in Table 5 – 16.

**Table 5 – 16 Velocity type vibration tester mandrel fitting
tolerance with the bearing bore**

Mandrel nominal dimension/mm		Mandrel tolerance/ μm	
over	to	upper	lower
3 ^①	18	– 9	– 15
18	30	– 12	– 18
30	50	– 14	– 21
50	80	– 17	– 25
80	120	– 23	– 32

① Include 3mm.

For acceleration type vibration tester, after combining the mandrel with the spindle, the radial runout where the mandrel is fitted with the inner ring is not larger than $5\mu\text{m}$, the axial runout of the mandrel with respect to the shaft abutment end face is not larger than $10\mu\text{m}$, the mandrel hardness should be 62 ~ 66 HRC. The deviation limit of the mandrel fitted with the bearing bore should be conformed with the values in Table 5 – 17.

**Table 5 – 17 Acceleration type vibration tester mandrel fitting
tolerance with the bearing bore**

Mandrel nominal dimension/mm		Mandrel deviation limit/ μm	
over	to	upper	lower
3 ^①	18	– 9	– 15
18	30	– 12	– 18
30	50	– 14	– 21
50	80	– 17	– 25
80	120	– 23	– 32

① Include 3mm.

4. Loading system

The loader for applying load to bearing outer ring, except for transmitting the constant load, and limiting the bearing outer ring rotation and the probably elastic recovering moment, can be the isolating system between the bearing and the machinery device, and make the bearing outer ring basically in free vibration condition.

(1) Axial direction loading

During the bearing measuring, for the deep groove ball bearing, the angular contact ball bearing and the tapered roller bearing, there should be applying certain combined axial direction load. The coaxiality between the combined axial direction load acting line and the driving spindle axial line can not be over $0.2\mu\text{m}$, as shown in Figure 5-3a. The angle between the combined axial direction load acting line and the driving spindle axial line can not be larger than 2° , as shown in Figure 5-3b.

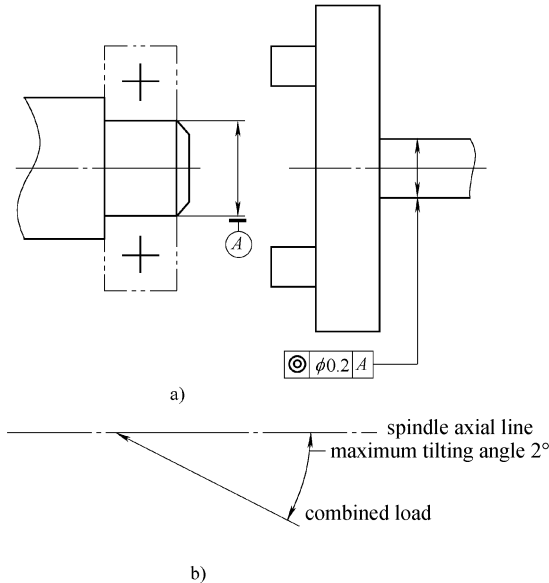


Figure 5-3 Combined axial direction load and spindle axial line coaxiality and tilting limit

a) coaxiality b) tilting limit

For velocity type vibration tester, the axial direction load should be conformed to the value of Table 5 – 18.

For acceleration type vibration tester, the axial direction load should be conformed to the value of Table 5 – 19.

Table 5 – 18 Velocity type vibration tester axial load and radial load

Bearing nominal bore diameter/mm		Axial load/N				Radial load/N
		deep groove ball bearing	angular contact ball bearing		tapered roller bearing	cylindrical roller bearing
$\alpha \leq 25^\circ$	$\alpha > 25^\circ$					
over	to					
3 ^①	6	20	—	—	—	—
6	9	30	—	—	—	—
9	20	40	60	100	60	150
20	30	80	110	160	110	
30	40					300
40	60	120	160	235	160	
60	80	180	235	350	235	600
80	120	225	340	440	340	

① Include 3mm.

Table 5 – 19 Acceleration type vibration tester axial load and radial load

Bearing nominal bore diameter/mm		Combined central axial load/N				Combined radial load/N
		deep groove ball bearing	angular contact ball bearing		tapered roller bearing	cylindrical roller bearing
over	to		$\alpha \leq 25^\circ$	$\alpha > 25^\circ$		
3	6	20	—	—	—	—
6	9	30	—	—	—	—
9	20	40	60	100	—	150
20 ^①	30	80	130	160	49	150
30	40	80	130	160	88	300
40	60	120	160	235	88	300
60	80	180	235	340	—	600
80	120	225	340	440	—	600

① Tapered roller bearing starting from 15mm.

(2) Radial loading

In measuring cylindrical roller bearing vibration, certain combined radial load should be applied on the bearing outer ring. For velocity type vibration tester, the radial direction load should be conformed to the value of Table 5 - 18. For acceleration type vibration tester, the radial direction load should be conformed to the value of Table 5 - 19.

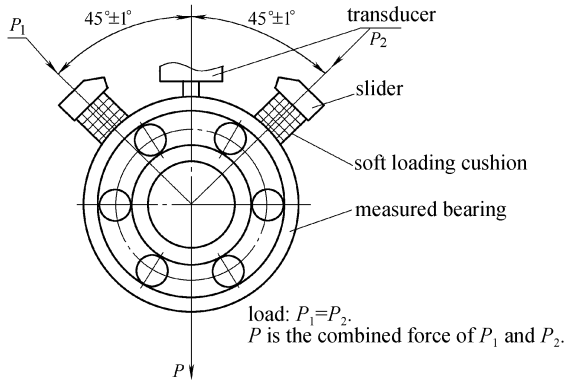


Figure 5 - 4 Radial loading sketch

The contact part between the radial loading cushion and the measured bearing outer ring should be as shown in Figure 5 - 4. The acting radial load should be vertical down, the angle between the radial load acting line and the vertical line through the driving spindle center can not be larger than 2° , the distance from the vertical line through the spindle center should be smaller than 0.5mm. The combined radial load acting line should be on the bearing outer ring geometric center $B/2$, as shown in Figure 5 - 5.

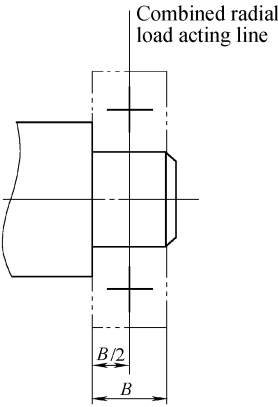


Figure 5 - 5 Combined radial load acting line

5. Transducer

(1) Velocity transducer

The transducer seat can be moved in the axial direction and the vertical direction respectively, and should guarantee the angle between the transducer contact load acting line on the measured bearing outer ring and the vertical line through the driving spindle center is not larger than 2° , and the distance from the vertical line through the spindle center should be smaller than 0.2mm.

The transducer induced signal is the variation rate of bearing outer ring radial vibration displacement. The transducer should not be separated from the measured bearing outer ring in the frequency range of 50 ~ 10000Hz, and it is should be guaranteed that the contact load acting on the measured bearing outer ring by the transducer is smaller than 0.7N.

The transducer system frequency respond should be within the limiting range in Figure 5-6.

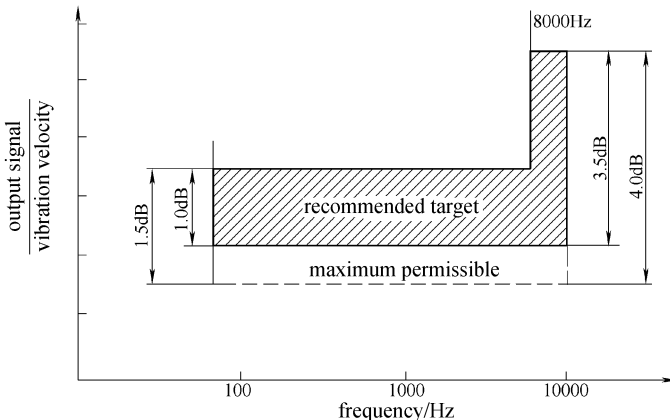


Figure 5-6 Velocity transducer frequency response characteristic

(2) Acceleration transducer

The coaxiality between the transducer transmission rod and the contacting load acting line and the vertical line through the bearing outer ring geometric center and perpendicular to the bearing axial line and the horizontal plane should not be over 0.2mm.

The contact load between the transducer transmission rod and the bearing outer ring cylindrical surface:

for bore diameter $d < 10\text{mm}$ is as 1.0 ~ 5.0N

for bore diameter $d \geq 10\text{mm}$ is as 5.0 ~ 10.0N

The piezoelectric transducer is used in the transducer system, the resonance frequency of the piezoelectric transducer should be larger than 20 kHz.

Within the measuring bearing vibration system acceleration range, the linear error of the acceleration transducer should be smaller than 5% .

The acceleration transducer system installation resonance frequency should be larger than 2000Hz.

Within 50 ~ 10000Hz frequency range, the acceleration transducer system tracing feature should be good , and the linear error of the amplitude should be smaller than 5% .

6. Electronic measuring device

(1) Velocity type vibration tester

The electronic measuring device should be within 50 ~ 10000Hz frequency resonance range, and divided as three 2.5 multiple frequency range wave filters, the wave filter band width should be conformed with the value in Table 5 – 20.

Table 5 – 20 Wave filter band width

Frequency	Low frequency band	Middle frequency band	High frequency band
frequency range	50 ~ 300	300 ~ 1800	1800 ~ 10000

The measuring time of the vibration velocity signal on every measuring point should be not less than 0.5 s, the reading should be read after the pointer being stable. If the signal oscillates, the middle value should be the reading.

The electronic measuring device wave filtering feature should be in the range in Figure 5 – 7, all the frequency attenuation of lower than 64% of the low cut-off frequency (f_L) and higher than 160% of the high cut-off frequency (f_H) should be not less than 40 dB.

The electronic measuring device should be examined in certain period, during the examination period, the permissible change range of the examining value can be as $\pm 4\%$. Figure 5 - 8 shows the velocity type transducer calibration system.

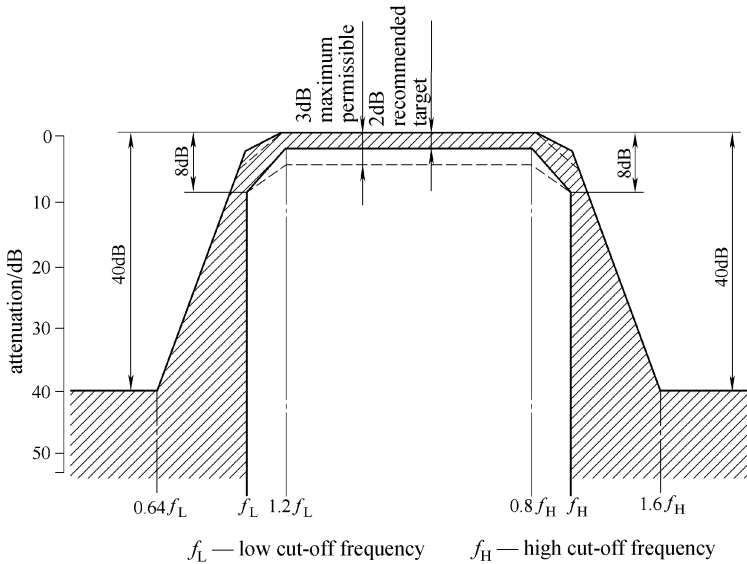


Figure 5 - 7 Wave filtering feature

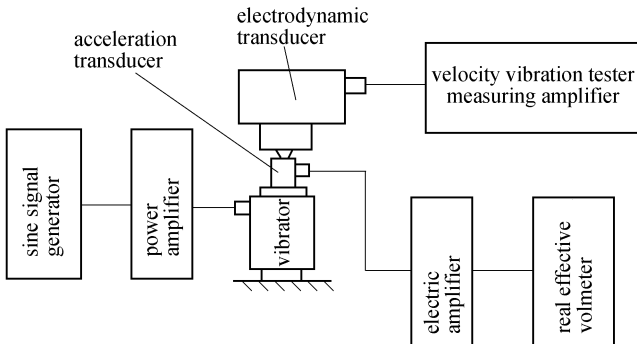


Figure 5 - 8 Velocity type transducer calibrating system

(2) Acceleration type transducer vibration tester

The electronic measuring device dynamic range should be not less than 60dB.

Within the electronic measuring device dynamic range, the base noise should be not larger than 5 dB. The electron measuring device through frequency band are as 50 ~ 10000Hz.

Within 50 ~ 10000Hz frequency range, the linear error of the combined amplitued of the electronic measuring device and the acceleration type transducer system should be less than 10%.

The electronic measuring device switch position error should be less than ± 0.30 dB, the indicating reading should be not larger than 0.5 dB.

Table 5 - 21 an Table 5 - 22 show the commonly used bearing vibration testers in China and other countries.

5.2.3 Rolling Bearing Vibration Standards

In order to improve Chinese rolling bearing quality, to reduce the rolling bearing noise level and to promote Chinese rolling bearing export, Chinese rolling bearing industry begins to make the rolling bearing vibration standards since the 1980's. At present, there are 11 rolling bearing vibration standards, in which including two measuring method standards, seven rolling bearing vibration standards and two steel ball standards as follows.

1. Rolling bearing vibration measuring standard

(1) JB/T 5313 — 2001 Rolling bearing vibration (velocity) measuring standard

This standard stipulated the bearing vibration (velocity) measuring method for inner diameter 3 ~ 120mm of the deep groove ball bearing, the angular contact ball bearing, the tapered roller bearing and the cylindrical roller bearing (N, NU, NJ and NF type). This standard is suitable for the above rolling bearing vibration (velocity) inspection in the laboratories, the bearing manufacturers and the users. The measured bearing vibration physical quantity is the radial vibration velocity on the bearing outer ring, which unit is $\mu\text{m/s}$;

Table 5 - 21 China produced bearing vibration tester

Tester type	Measured bearing type	Measuring range	Measuring physical quantity	Measuring unit	Frequency band/Hz	Rotation speed/(r/min)	Loading	Maker
BVT-1	deep groove ball bearing angular contact ball bearing Tapered roller bearing	inner diameter $\phi 5 \sim$ outer diameter $\phi 170$	velocity	$\mu\text{m/s}$	low frequency $50 \sim 300$ middle frequency $300 \sim 1800$ high frequency $1800 \sim 10000$	1800	axial	hangzhou bearing test and research center
BVT-2A	cylindrical roller bearing needle roller bearing	inner diameter $\phi 15 \sim \phi 60$				1800	radial	
BVT-3A	cylindrical roller bearing needle roller bearing	inner diameter $> \phi 60$ outer diameter $\leq \phi 280$				900	radial	
BVT-4	deep groove ball bearing angular contact ball bearing Tapered roller bearing	inner diameter $> \phi 60 \sim$ outer diameter $\leq \phi 280$				900	axial	
BVT-5	deep groove ball bearing angular contact ball bearing	inner diameter $> \phi 15 \sim \phi 60$				1800	axial radial	
BVT-6	tapered roller bearing cylindrical roller bearing	inner diameter $> \phi 60 \sim$ outer diameter $\leq \phi 280$				900	axial radial	
S0910	deep groove ball bearing angular contact ball bearing	inner diameter $> \phi 10 \sim \phi 70$	acceleration	dB	250 ~ 10000	1440	axial	dalian bearing yiqi plant

Table 5 – 22 Other country produced bearing vibration tester

Tester type	Measured bearing type	Measuring bearing range/mm	Rotation speed / (r/min)	Loading	Measuring physical quantity	Measuring unit	Frequency band	Maker
B1010	deep groove ball bearing, angular contact ball bearing, tapered roller bearing	inner diameter $\phi 8 \sim \phi 70$ outer diameter $\phi 25.4 \sim \phi 152$	1800	axial	ANDERON	ANDERON	low frequency 50 ~ 300 Hz middle frequency 300 ~ 1.8 kHz high frequency 1.8 ~ 10 kHz	Bendix USA
AD-MN-4		inner diameter $\geq \phi 4$ outer diameter $\leq \phi 170$	1800	axial	ANDERON	ANDERON		Japan Jian Yuan
AD-SN-4		inner diameter $\geq \phi 0.8$ outer diameter $\leq \phi 25$	1800	axial	ANDERON	ANDERON		
MGG11		inner diameter $\geq \phi 3$ outer diameter $\leq \phi 100$	1800	axial	velocity	$\mu\text{m/s}$		FAG Germany
MVH-90B		outer diameter $\phi 26 \sim \phi 90$	1800	axial	velocity	$\mu\text{m/s}$		SKF Sweden
MVH-200A		outer diameter $\phi 90 \sim \phi 200$	700	axial	velocity	$\mu\text{m/s}$		

Within the 50 ~ 10000Hz frequency range, the three bearing vibration frequency band should be conformed with the values of Table 5 – 20.

The base vibration of the measuring device should be conformed with the values of Table 5 – 12.

During the measuring, the bearing inner ring actual rotation speed should be conformed with the values of Table 5 – 14.

The mandrel hardness should be 61 ~ 64 HRC.

The mandrel fit tolerance with the bearing bore should be conformed with the values of Table 5 – 16.

The axial load or radial load applied in the measuring process should be conformed with the values of Table 5 – 18.

The transducer frequency response should be conformed with the values of Table 5 – 6.

The electronic measuring device wave filtering feature should be within the stipulated range in Table 5 – 7 and the electronic measuring device should be examined in certain period.

Besides, this standard also stipulated that the filling grease bearing can be measured in the filling grease condition. For ordinary bearing, it is the first step to wash the bearing and then filling the N15 machinery oil after the washing agent is fully volatilized. The kinematic viscosity at 40℃ should be 13.5 ~ 16.5mm²/s.

(2) JB/T 5314 — 2002 Rolling bearing vibration (acceleration) measuring standard

This standard stipulated the bearing vibration (acceleration) measuring method for inner diameter 3 ~ 120mm of the deep groove ball bearing, the angular contact ball bearing, the tapered roller bearing and the cylindrical roller bearing (N, NF, NU, NJ type). The unit of the vibration acceleration degree, the base vibration acceleration degree and the bearing vibration degree peak value is all expressed by decibel (dB). The zero dB corresponds to one thousandth of the earth gravity acceleration. The calculation equation of dB value is listed in Table 5 – 10.

The base vibration of the measuring device should be conformed with the values of Table 5 – 13.

The mandrel hardness should be 62 ~ 66 HRC.

The mandrel fit tolerance with the bearing bore should be conformed with the values of Table 5 – 17.

The spindle rotation speed should be conformed with the value of Table 5 – 15.

The axial load or the radial load applied in the measuring process should be conformed with the value of Table 5 – 19.

The transmission vibration bar contacting load with the bearing outside cylindrical surface: for bore diameter $d < 10\text{mm}$, is 1.0 ~ 5.0N; for bore diameter $d \geq 10\text{mm}$, is 5.0 ~ 10.0N.

The used acceleration transducer resonance frequency should be larger than 20 kHz;

Within the measured bearing vibration acceleration degree range, the linear error of the acceleration transducer amplitude should be less than 5%.

The electronic measuring device base noise should be not larger than 5 dB, and the through frequency band is as 50 ~ 10000Hz. Within this frequency band range, the combined linear error of the electronic measuring device and the transducer system should be less than 10%, the electronic measuring device switch error should be less than ± 0.3 dB, and the indicated reading should be not larger than 0.5 dB. The measuring instrument should be calibrated once a year.

Besides, this standard also stipulated that the once filling grease bearing (such as double side sealing bearing) can be measured in sealing condition, without washing and re-lubrication. For not-once filling lubrication bearing, it is the first step to wash the bearing by proper washing agent, under the measuring environment temperature filling the N32 (environment temperature 10 ~ 20°C) or N46 (environment temperature 20 ~ 30°C) lubrication oil. Their kinematic viscosity should be within 80 ~ 150mm²/s.

When single side measuring, the average value of the three measured bearing points is the bearing acceleration degree. When double side measuring, the largest value of the average value of the three measured bearing points is the bearing acceleration degree.

2. Rolling bearing vibration measuring standard

Till present, five rolling bearing vibration standards have been made and put into application. Table 5 - 23 shows the application range, the standard number, the measuring unit, the application dimension range and the measuring frequency band.

(1) Deep groove ball bearing vibration (velocity) standard JB/T 10187 — 2000

This standard is suitable for the deep groove ball bearings of the bore diameter 3 ~ 120mm, and 0 group radial clearance. The stipulated bearing vibration (velocity) V group is suitable for the bearing manufacturer production inspection, the V_1 , V_2 , V_3 , V_4 groups are suitable for the bearings that the user has requirement to control the bearing vibration level, and can be used as the bearing manufacturer production inspection and the user inspection.

The measuring unit is $\mu\text{m/s}$ (RMS).

For the bore diameter 3 ~ 60mm deep groove ball bearing of the 0, 2, 3 series, the single bearing vibration (velocity) value should be conformed with the stipulated value in Table 5 - 24. The bearing vibration measuring of the bore diameter 10 ~ 120mm should be conformed with the JB /T5313 stipulation.

For the deep groove ball bearing with the bore diameter 65 ~ 120mm of the 0, 2, 3 series, the single bearing vibration (velocity) value should be conformed with the stipulated value in Table 5 - 25.

The miniature deep groove ball bearing measuring method with the bore diameter 3 ~ 9mm is: while the driving device starting, the transducer mounting position vibration should be conformed with the stipulation in Table 5 - 26.

Table 5 – 23 Rolling bearing vibration standard^①

Bearing type	Deep groove ball bearing		Cylindrical roller bearing	Tapered roller bearing	
standard number	JB/T 10187—2000	JB/T 7047—1999	JB/T 8922—1999	JB/T 10236—2001	JB/T 10237—2001
measuring physical quality	velocity	acceleration	velocity	velocity	acceleration
measuring unit	$\mu\text{m/s}$	dB	$\mu\text{m/s}$	$\mu\text{m/s}$	dB
bearing dimension range	$\phi 3 \sim \phi 120$	$\phi 3 \sim \phi 120$	$\phi 15 \sim \phi 120$	$\phi 15 \sim \phi 60$	$\phi 15 \sim \phi 60$
measuring frequency band/Hz	low frequency 50 ~ 300 middle frequency 300 ~ 1800 high frequency 1800 ~ 10000	through frequency band 250 ~ 10000	low frequency 50 ~ 300 middle frequency 300 ~ 1800 high frequency 1800 ~ 10000	low frequency 50 ~ 300 middle frequency 300 ~ 1800 high frequency 1800 ~ 10000	through frequency band 250 ~ 10000
measuring condition	Table 5 – 18, Table 5 – 27	Table 5 – 19, Table 5 – 30	Table 5 – 18	Table 5 – 18	Table 5 – 34
vibration standard value	Table 5 – 24, Table 5 – 25	Table 5 – 28, Table 5 – 29	Table 5 – 31, Table 5 – 32	Table 5 – 33	Table 5 – 35

① The standard bearing vibration value designates that the bearing radial vibration effective value (RMS) is on the intersection point of through the raceway center section with the outer ring outside cylindrical surface, the bearing inner ring is rotating, and the outer ring is stationary.

Table 5-24 Single bearing vibration (velocity) limit value (1)(unit: $\mu\text{m/s}$)

Nominal bearing bore diameter d/mm	$V^{\text{①}}$			V1			V2			V3			V4		
	low ^②	mid ^②	high ^②	low	mid	high	low	mid	high	low	mid	high	low	mid	high
3	80	44	44	60	35	32	48	26	22	31	16	15	28	10	10
4	80	44	44	60	35	32	48	26	22	31	16	15	28	10	10
5	110	72	60	74	48	40	58	36	30	35	21	18	32	11	11
6	110	72	60	74	48	40	58	36	30	35	21	18	32	11	11
7	130	96	80	92	66	54	72	48	40	44	28	24	38	12	12
8	130	96	80	92	66	54	72	48	40	44	28	24	38	12	12
9	130	96	80	92	66	54	72	48	40	44	28	24	38	12	12
10	160	120	100	120	80	70	90	60	50	55	35	30	45	14	15
12	160	120	100	120	80	70	90	60	50	55	35	30	45	14	15
15	210	150	120	150	100	85	110	78	60	65	46	35	52	18	18
17	210	150	120	150	100	85	110	78	60	65	46	35	52	25	25
20	260	190	150	180	125	100	130	100	75	80	60	45	60	25	25

(continued)

Nominal bearing bore diameter d/mm	V ^①			V1			V2			V3			V4		
	low ^②	mid ^②	high ^②	low	mid	high	low	mid	high	low	mid	high	low	mid	high
22	260	190	150	180	125	100	130	100	75	80	60	45	60	30	32
25	260	190	150	180	125	100	130	100	75	80	60	45	60	30	32
28	260	190	150	180	125	100	130	100	75	80	60	45	60	35	40
30	300	240	190	200	150	130	150	120	100	90	75	60	70	35	40
32	300	240	190	200	150	130	150	120	100	90	75	60	70	35	40
35	300	240	190	200	150	130	150	120	100	90	75	60	70	42	45
40	360	300	260	240	180	160	180	150	130	110	90	80	82	50	50
45	360	300	260	240	180	160	180	150	130	110	90	80	82	60	60
50	420	320	320	280	200	200	210	160	160	125	100	100	95	70	70
55	420	360	360	280	220	200	210	180	180	125	110	110	95	70	70
60	480	360	440	320	220	240	240	180	200	145	110	130	100	80	80

① not expressed in the bearing model.

② low frequency, middle frequency, high frequency.

Table 5-25 Single bearing vibration (velocity) limit value (2)(unit: $\mu\text{m/s}$)

Nominal bearing bore diameter d/mm	$V^{\text{①}}$			V1			V2			V3			V4		
	low ^②	mid ^②	high ^②	low	mid	high	low	mid	high	low	mid	high	low	mid	high
65	300	260	420	180	160	240	130	100	150	105	80	105	50	50	75
70	360	310	460	200	180	280	150	120	200	110	90	135	58	58	88
75	360	310	460	200	180	280	150	120	200	110	90	135	58	58	88
80	420	360	540	240	210	320	180	120	240	130	110	160	65	65	100
85	420	360	540	240	210	320	180	150	240	130	110	160	65	65	100
90	480	420	600	290	250	370	210	180	270	145	125	180	75	75	115
95	480	420	600	290	250	370	210	180	270	145	125	180	75	75	115
100	560	490	670	340	300	420	250	215	310	170	145	200	88	88	135
105	560	490	670	340	300	420	250	215	310	170	145	200	88	88	135
110	640	570	750	400	350	480	290	260	350	190	175	225	100	100	160
120	640	570	750	400	350	480	290	260	350	190	175	225	100	100	160

① not expressed in the bearing model.

② low frequency, middle frequency, high frequency.

When measuring the miniature ball bearing vibration, certain axial load should be conformed with the stipulation in Table 5 – 27. The fit tolerance of the mandrel with the measured bearing bore is 10mm. The spindle rotation speed is 1800r/min.

Table 5 – 26 Miniature deep groove ball bearing base vibration

Frequency range	50 ~ 300Hz	> 300 ~ 1800Hz	> 1800 ~ 10000Hz
vibration/ ($\mu\text{m/s}$)	≤ 12	≤ 7	≤ 5

Table 5 – 27 Miniature deep groove ball bearing axial load

Nominal bearing bore diameter d/mm	axial load
3	20
4	
5	
6	
7	30
8	
9	

If any frequency band vibration (velocity) value of a single bearing is larger than the value in Table 5 – 24 or Table 5 – 25, this bearing is judged as the non-qualified bearing.

The sealed bearing vibration (velocity) value is the measuring value with filling grease and with seals or with shields.

(2) Deep groove ball bearing vibration (acceleration) standard JB /T 7047 — 1999

This standard is suitable for the deep groove ball bearings of the bore diameter 3 ~ 120mm, and 0 group radial clearance. The stipulated bearing vibration (acceleration) Z group is the basic requirement for common bearings, which is suitable for the bearing manufacturer production inspection, and there is no need to mark “ Z ” on the bearings. The Z_1 , Z_2 , Z_3 , Z_4 groups are suitable for the bearing manufacturer production inspection and the user acceptance. The measuring condition should be conformed with the stipulation in JB/T5314.

For the deep groove ball bearing of the 0, 2, 3 series with the bore

diameter 3 ~ 60mm, the single bearing vibration (acceleration degree) value should be conformed with the stipulated value in Table 5 – 28. For the deep groove ball bearing of the 0, 2, 3 diameter series with the bore diameter 65 ~ 120mm, the single bearing vibration acceleration degree should be conformed with the stipulated value in Table 5 – 29. For the bearings with seals or with shields, the measuring should be performed after mounting the seals or the shields.

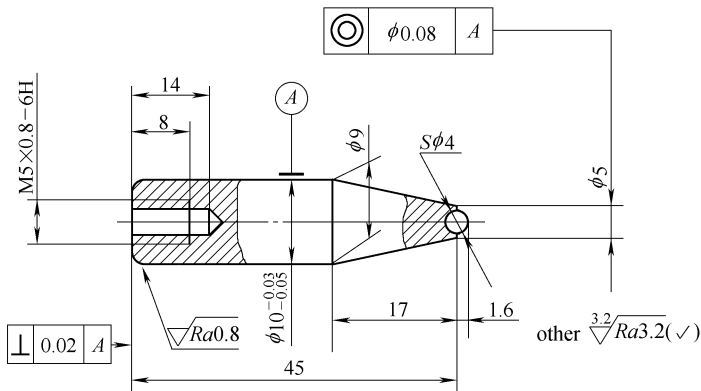
The single bearing (designates any single bearing in the sample), if its vibration degree is over the stipulated value in Table 5 – 28, Table 5 – 29, this bearing is judged as the non-qualified bearing.

For the miniature deep groove ball bearing with the bore diameter 3 ~ 9mm, the measuring method is that the spindle rotation speed is 1500r/min $\pm$ 30r/min.

The tolerance with the bore diameter is f_5 .

The axial load acted on the measured bearing outer ring should be conformed with the value in Table 5 – 30. The error is $\pm 3N$;

The transmission vibration bar mass should be 21g $\pm$ 1g. The transmission vibration bar geometry and dimension should be conformed with the stipulation in Figure 5 – 9.



transmission vibration bar material: GCr 15

Sφ4: YG series hard alloy (Hardness should be not less than 84.5 HRA, surface roughness Ra should be not larger than 0.125 μ m).

Figure 5 – 9 Transmission vibration bar dimension

Table 5-28 Deep groove ball bearing acceleration degree value (1)

Bearing nominal bore diameter/mm	Single bearing vibration acceleration degree (dB) ($\leq$)														
	diameter series(0)					diameter series(2)					diameter series(3)				
	Z	Z1	Z2	Z3	Z4	Z	Z1	Z2	Z3	Z4	Z	Z1	Z2	Z3	Z4
3	35	34	32	28	24	36	35	32	30	—	37	36	33	31	—
4	35	34	32	28	24	36	35	32	30	—	37	36	33	31	—
5	37	36	34	30	26	38	37	34	32	—	39	37	35	33	—
6	37	36	34	30	26	38	37	34	32	—	39	37	35	33	—
7	39	38	35	31	27	40	38	36	34	—	—	—	—	—	—
8	39	38	35	31	27	40	38	36	34	—	—	—	—	—	—
9	41	40	36	32	28	42	40	37	35	—	—	—	—	—	—
10	43	42	38	33	28	44	42	39	35	30	46	44	40	37	32
12	44	43	39	34	30	45	43	39	35	30	47	45	40	37	32
15	45	44	40	35	30	46	44	41	36	31	48	46	42	38	33
17	46	44	40	35	30	47	45	41	36	31	49	47	42	38	33
20	47	45	41	36	31	48	46	42	38	33	50	48	43	39	34

(continued)

Bearing nominal bore diameter/mm	Single bearing vibration acceleration degree (dB) ($\leq$)														
	diameter series(0)					diameter series(2)					diameter series(3)				
	Z	Z1	Z2	Z3	Z4	Z	Z1	Z2	Z3	Z4	Z	Z1	Z2	Z3	Z4
22	47	45	41	36	31	48	46	42	38	33	50	48	43	39	34
25	48	46	42	38	34	49	47	43	40	36	51	49	44	41	37
28	49	47	43	39	35	50	48	44	41	37	52	50	45	42	38
30	49	47	43	39	35	50	48	44	41	37	52	50	45	42	38
32	50	48	44	40	36	51	49	45	42	38	53	51	46	43	39
35	51	49	45	41	37	52	50	46	43	39	54	52	47	44	40
40	53	51	46	42	38	54	52	47	44	40	56	54	49	45	41
45	55	53	48	45	42	56	54	49	46	43	58	56	51	47	44
50	57	54	50	47	63	58	55	51	48	45	60	57	53	49	46
55	59	56	52	49	65	60	57	53	50	47	62	59	54	51	48
60	61	58	54	51	67	62	59	54	51	48	64	61	56	53	50

Table 5 - 29 Deep groove ball bearing acceleration degree value (2)

Bearing nominal bore diameter/mm	Single bearing vibration acceleration degree (dB) (≤)											
	diameter series(0)				diameter series(2)				diameter series(3)			
	Z	Z1	Z2	Z3	Z	Z1	Z2	Z3	Z	Z1	Z2	Z3
65	49	48	46	41	50	49	47	42	51	50	48	43
70	50	49	47	42	51	50	48	43	52	51	49	44
75	51	50	48	43	52	51	49	44	53	52	50	45
80	52	51	49	44	53	52	50	45	54	53	51	46
85	53	52	50	45	54	53	51	46	56	55	52	47
90	54	53	52	47	56	55	53	48	58	57	54	49
95	56	55	54	49	58	57	55	50	60	59	56	51
100	58	57	56	51	60	59	57	52	62	61	58	53
105	60	59	58	53	62	61	59	54	64	63	60	55
110	62	61	60	55	64	63	61	56	66	65	62	57
120	64	63	62	57	66	65	63	58	68	67	64	59

Table 5 – 30 Miniature deep groove ball bearing axial load

Nominal bearing bore diameter d /mm	combined axial load/N
3	30
4	
5	
6	
7	40
8	
9	

(3) Cylindrical roller bearing vibration (velocity) standard JB/T 8922 — 2011

This standard is suitable for the vibration (velocity) of the cylindrical roller bearings: the nominal bore diameter are 15 ~ 120mm, the bearing types are the NU type, the NJ type, the N type and the NF type, the diameter series are the 02 series, the 03 series, and the radial clearance is 0 group. The V group is suitable for the manufacturer inspection, the groups of V1, V2, V3 are suitable for the bearings that the user has the requirement to control the bearing vibration level and for the user acceptance. The measuring method is that according to the stipulation in JB/T 5313 — 2001, during measuring, the acting radial load should be conformed with the value in Table 5 – 18.

The single bearing vibration (velocity) value should be conformed with the stipulated value in Table 5 – 31 and Table 5 – 32.

If any frequency band vibration (velocity) value of a single bearing is larger than the value in Table 5 – 31 or Table 5 – 32, this bearing is judged as the non-qualified bearing.

(4) Tapered roller bearing vibration (velocity) standard JB/T 10236 — 2001

This standard is suitable for the vibration (velocity) of the tapered roller bearings: the nominal bore diameter are 15 ~ 60mm, the diameter

Table 5 – 31 Cylindrical roller bearing single bearing vibration (velocity) limit value (1) (unit: $\mu\text{m/s}$)

Nominal bearing bore diameter d/mm	$V^{①}$			V1			V2			V3		
	low	mid	high	low	mid	high	low	mid	high	low	mid	high
15	340	420	420	260	310	310	200	190	190	140	100	100
17	370	460	460	290	350	350	230	220	220	160	110	110
20	370	460	460	290	350	350	230	220	220	160	110	110
25	420	530	530	330	400	400	260	260	260	180	130	130
30	420	530	530	330	400	400	260	260	260	180	130	130
35	490	610	610	380	470	470	300	300	300	210	150	150
40	490	610	610	380	470	470	300	300	300	210	150	150
45	570	690	690	430	540	540	340	340	340	240	170	170
50	570	690	690	430	540	540	340	340	340	240	170	170
55	650	780	780	500	610	610	400	380	380	280	190	190
60	650	780	780	500	610	610	400	380	380	280	190	190

① not expressed in the bearing model.

Table 5 - 32 Cylindrical roller bearing single bearing vibration (velocity) limit value (2) (unit: $\mu\text{m/s}$)

Nominal bearing bore diameter d/mm	$V^{①}$			V1			V2			V3		
	low ^②	mid ^②	high ^②	low	mid	high	low	mid	high	low	mid	high
65	420	500	500	310	380	380	240	230	230	170	120	120
70	470	560	560	350	430	430	290	270	270	200	140	140
75	470	560	560	350	430	430	290	270	270	200	140	140
80	530	630	630	410	500	500	330	300	300	230	160	160
85	530	630	630	410	500	500	330	300	300	230	160	160
90	610	710	710	460	570	570	370	350	350	260	180	180
95	610	710	710	460	570	570	370	350	350	260	180	180
100	690	800	800	540	650	650	430	400	400	300	210	210
105	690	800	800	540	650	650	430	400	400	300	210	210
110	780	920	920	630	740	740	500	470	470	350	240	240
120	780	920	920	630	740	740	500	470	470	350	240	240

① not expressed in the bearing model.

② low frequency, middle frequency, high frequency.

Table 5 - 33 Cylindrical roller bearing single bearing vibration (velocity) limit value (unit: $\mu\text{m/s}$)

Nominal bearing bore diameter d/mm	$V^{①}$			V1			V2			V3		
	low	mid	high	low	mid	high	low	mid	high	low	mid	high
15	310	500	500	220	360	360	150	220	220	100	100	100
17	330	550	550	240	400	400	170	240	240	110	110	110
20	330	550	550	240	400	400	170	240	240	110	110	110
25	360	590	600	280	440	450	210	280	280	120	140	130
30	360	590	600	280	440	450	210	280	280	120	140	130
35	400	640	670	320	480	500	250	320	300	150	180	160
40	440	690	740	360	530	560	280	350	320	170	210	190
45	440	690	740	360	530	560	280	350	320	170	210	190
50	480	750	810	400	600	620	320	400	360	220	260	240
55	480	750	810	400	600	680	320	400	360	220	260	240
60	530	850	1000	450	680	760	370	460	420	300	330	300

① not expressed in the bearing model.

series are the 30200 series, the 32200 series, the 30300 series and the 32300 series. The V group is suitable for the manufacturer inspection, the groups of V1, V2, V3 are suitable for the bearings that the user has the requirement to control the bearing vibration level and for the user acceptance. The tapered roller bearing vibration (velocity) measuring method is according to the stipulation in JB/T 5313 — 2001.

If any frequency band vibration (velocity) value of a single bearing is larger than the value in Table 5 – 33, this bearing is judged as the non-qualified bearing.

(5) Tapered roller bearing vibration (acceleration) standard JB/T 10237 — 2001

This standard is suitable for the vibration (velocity) of the tapered roller bearings: the nominal bore diameter are 15 ~ 60mm, the diameter series are the 30200 series, the 32200 series, the 30300 series and the 32300 series. The vibration acceleration Z group is the basic requirement, suitable for the manufacturer inspection, the groups of Z1, Z2, Z3 are suitable for the bearings that the user has the requirement to control the bearing vibration acceleration level and for the manufacturer inspection and for the user acceptance. The driving device base vibration should be not larger than 15 dB. The acting axial load should be conformed with the value in Table 5 – 34. The tapered roller bearing vibration (acceleration) measuring method is according to the stipulation in JB/T 5314 — 2001.

If the single bearing vibration acceleration degree is larger than the value in Table 5 – 35, this bearing is judged as the non-qualified bearing.

Table 5 – 34 Tapered roller bearing axial load

Nominal bearing bore diameter d /mm		Acting axial load /N
over	to	
15 ^①	30	49 ± 4.9
30	60	88 ± 4.9

① include 15mm.

Table 5 – 35 Tapered roller bearing vibration (acceleration)
degree value (dB)

Nominal bearing bore diameter d/mm	30200 series , 32200 series			30300 series , 32300 series		
	Z ^①	Z1	Z2	Z	Z1	Z2
15	—	—	—	56	54	50
17	56	54	50	58	56	52
20	57	55	51	61	58	53
25	58	56	54	64	61	56
30	59	56	52	67	64	59
35	51	58	53	68	65	60
40	63	60	55	69	66	61
45	65	62	57	69	66	61
50	67	64	59	71	68	63
55	69	66	61	74	71	66
60	71	68	63	77	74	69

① not expressed in the bearing model.

3. Steel ball vibration standard

The most active component in the rolling bearing is the ball , therefore the ball is the most important factor for influencing the ball bearing vibration and noise. In order to control the bearing vibration and noise level , the ball quality should be controlled. In China , the ball vibration measuring tester has been developed and the related standard also has been made.

(1) Ball vibration (acceleration) standard JB/T 8923 — 1999

This standard is suitable for the commercialized ball vibration acceleration that the ball diameter is within 3 ~ 30.1625mm and the tolerance degree is not lower than the G20 degree.

The ball vibration designates the vibration that was caused by the radial displacement because of the ball surface deviating the theoretical spherical position while the ball is rotating about the ball center line.

The vibration acceleration degree designates the value that the commonly used logarithmic of the ratio of the square root of the vibration acceleration value within one frequency band range to the reference acceleration value and then times twenty and then adds one constant, as shown in the following equation:

When the ball diameter is as 3 ~ 15.5mm, the ball vibration acceleration degree is as:

$$L = 20 \log \frac{a}{g} + 96$$

When the ball diameter is as 15.5 ~ 30.1625mm, the ball vibration acceleration degree is as:

$$L = 20 \log \frac{a}{g} + 95$$

Where L — ball vibration acceleration degree (dB);

g — reference acceleration value, $g = 9.81 \text{ m/s}^2$;

a — measured vibration acceleration value (m/s^2).

The ball vibration acceleration degree designates the vibration acceleration degree on the maximum circle that is perpendicular to the spherical center line of the ball while the ball rotates about the spherical center line.

The ball vibration acceleration degree measuring tester, the measuring principle, the measuring method and the measuring environment should be conformed with the requirement in JB/T 8075. Before measuring, it is the first step to wash the ball in the NY-120 solvent. After washing, there should be no foreign matter such as the dust, the wearing particles on the ball surface. During measuring, the ball reading value should be as the middle value within the indicated range after the indicator stably swings for 5s. Every ball should be measured in three different sections, taking the maximum reading value as that ball vibration value.

For the average value of the ball lot vibration measuring: for the ball lot that the ball diameter is as 3 ~ 15.5mm, the measurement should be performed for random sampling of twenty balls per lot, taking the vibration average value as this ball lot vibration average value; for the ball lot that the

ball diameter is as 15.5 ~ 20mm, the measurement should be performed for random sampling of ten balls per lot, taking the vibration average value as this ball lot vibration average value; for the ball lot that the ball diameter is as 20 ~ 30mm, the measurement should be performed for random sampling of five balls per lot, taking the vibration average value as this ball lot vibration average value.

There are three groups for ball vibration acceleration degree: Z1, Z2, Z3. The ball diameter is equal to or smaller than 15.5mm, the ball vibration acceleration degree should be conformed with the stipulation value in Table 5 – 36 and Table 5 – 37. The ball diameter is larger than 15.5mm, the ball vibration acceleration degree should be conformed with the stipulation value in Table 5 – 38 and Table 5 – 39.

Table 5 – 36 Ball vibration acceleration degree (1)

(unit: dB)

Nominal ball diameter /mm	Ball vibration acceleration degree max					
	Z1		Z2		Z3	
	lot average	single	lot average	single	lot average	single
>3 ~ 4	36.0	38.0	34.5	36.5	33.0	35.0
>4 ~ 5	36.5	38.5	35.0	37.0	33.5	35.5
>5 ~ 6	37.0	39.0	35.5	37.5	34.0	36.0
>6 ~ 7	37.5	39.5	36.0	38.0	34.5	36.5
>7 ~ 8	38.0	40.0	36.5	38.5	35.0	37.0
>8 ~ 9	38.5	40.5	37.0	39.0	35.5	37.5
>9 ~ 10	39.0	41.0	37.5	39.5	36.0	38.0
>10 ~ 11	39.5	41.5	38.0	40.0	36.5	38.5
>11 ~ 12	40.0	42.0	38.5	40.5	37.0	39.0
>12 ~ 13	40.5	42.5	39.0	41.0	37.5	39.5
>13 ~ 14	41.0	43.0	39.5	41.5	38.0	40.0
>14 ~ 15.5	41.5	43.5	40.0	42.0	38.5	40.5

If the ball lot vibration average value is not over the stipulation value in Table 5 – 36 and Table 5 – 38 , but a single maximum vibration value is over, then the exceeded difference value should be added to that lot vibration average value taking this value as the new judging value. After adding the exceeded difference value, if the value is still not over the stipulation value in Table 5 – 36 and Table 5 – 38 and at the same time , this lot value is also met the stipulation value in Table 5 – 37 and Table 5 – 39 , then this lot value is judged as qualified. If this lot value is over the stipulation value in Table 5 – 37 and Table 5 – 39 , then this lot value is judged as not qualified and the tolerance degree of this lot of balls should be reduced.

Table 5 – 37 Ball lot vibration acceleration limit value difference (1)

(unit: dB)

Maximum difference between the maximum value and the minimum value in a ball lot	Z1	Z2	Z3
	6	5	4

Table 5 – 38 Ball vibration acceleration degree (2)

(unit: dB)

Nominal ball diameter /mm	Ball vibration acceleration degree max					
	Z1		Z2		Z3	
	lot average	single	lot average	single	lot average	single
> 15. 5 ~ 16	57	59	55	57	53	55
> 16 ~ 18	58	60	56	58	54	56
> 18 ~ 20	59	61	57	59	55	57
> 20 ~ 22	60	62	58	60	56	58
> 22 ~ 24	61	63	59	61	57	59
> 24 ~ 26	62	64	60	62	58	60
> 26 ~ 28	63	65	61	63	59	61
> 28 ~ 30. 1625	64	66	62	64	60	62

Table 5 – 39 Ball lot vibration acceleration limit value difference (2)

(unit: dB)

Maximum difference between the maximum value and the minimum value in a ball lot	Z1	Z2	Z3
	7	6	5

(2) Ball vibration (velocity) recommended value

Suitable for the vibration velocity of the commercialized ball that the diameter is 3 ~ 22.225mm, and the tolerance that is not lower than G20 degree. This ball vibration (velocity) recommended value can be used as the reference for the manufacturer inspection and the user acceptance for the balls that the user has vibration requirement for the balls of the ball bearing.

Measuring method: The ball vibration designates the vibration causing the radial displacement while the ball is rotating about any ball diameter because of the ball surface deviating the theoretical spherical geometry to let the spherical surface instantaneous position deviating the theoretical spherical position. When the ball is rotating, the root mean-square of the radial vibration velocity value on the ball surface maximum circle perpendicular to the rotating diameter is as the ball vibration velocity value, which unit is $\mu\text{m/s}$.

Before measuring, the balls should be washed in the NY-120 gasoline. After washing, there should be no dust or wearing particles and foreign matter on the ball surface. After the solvent has been fully evaporated, in order to lubricate the ball surface, the clean N15 machinery oil should be added in (the kinematic viscosity is $13.5 \sim 16.5\text{mm}^2/\text{s}$ at 40°C). When mounting the ball, the ball should be pressed closely to the inner tapered surface of the mandrel, and then pressing the ball tightly along the axial direction by the pressing plate. While the ball diameter is smaller than 15.5mm, the pressing force is 30N; while the ball diameter is larger than 15.5mm and smaller than 30.162mm, the pressing force is 40N. Adjusting the position of the measuring tip of the transducer such that the measuring tip of the transducer is located on the upper position of the maximum circle

that is perpendicular to the ball rotating axial line, the measuring force should be $0.4 \sim 0.55\text{N}$. Then selecting the appropriate measuring range, the measuring time on every measuring section should be not more than 2 s , while the indicator is stable, the middle reading within the indicated value range should be the measuring value of this section. The measurement should be performed in three different sections and then the maximum measuring value within one frequency band should be as the vibration velocity value of this frequency band.

Measuring technical requirements:

Rotation speed: rotation speed of ball rotating about diameter is $1800\text{r/min} \pm 36\text{r/min}$.

Frequency band: low frequency band: $150 \sim 900\text{Hz}$

high frequency band: $900 \sim 5400\text{Hz}$

Base vibration: A cylindrical measuring wick bar which diameter is larger than 50mm and the parallelism of the upper plane and the lower plane is $\leq 10\mu\text{m}$ places between the transducer measuring tip and the supporting plate of the measuring instrument spindle. Then the measuring tip of the transducer is pressed onto the end face of the measuring wick bar in the measuring condition, and the spindle is started. The measured base vibration without ball should be as: the low frequency value can not be larger than $4\mu\text{m/s}$; the high frequency value can not be larger than $5\mu\text{m/s}$.

Measuring transducer: measuring the radial displacement varying rate of the ball surface by the velocity type transducer. Within the frequency range $150 \sim 5400\text{Hz}$, the transducer deviation should not be produced and it should be guaranteed that the contact load is less than 0.55N . The transducer measuring tip radius is 0.6mm , and the transducer frequency resonance should be in the limit range in Figure 5 – 10.

Electronic measuring device: The electronic measuring device frequency resonance range should be $150 \sim 5400\text{Hz}$, and dividing as two 2.5 double frequency range filters, their band widths are as: low frequency band (L): $150 \sim 900\text{Hz}$; high frequency band (H): $900 \sim 5400\text{Hz}$.

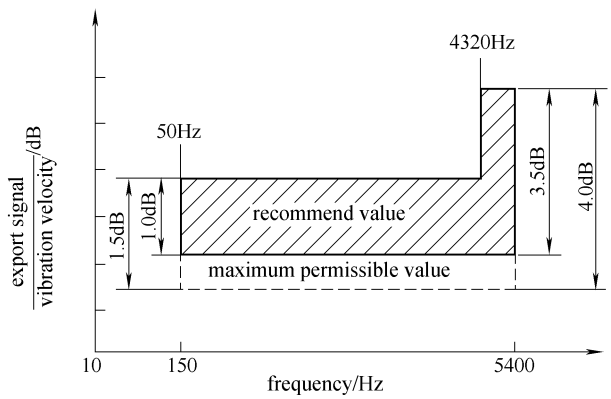


Figure 5-10 Transducer frequency resonance feature

The electronic measuring device filtering feature should be in the range shown in Figure 5-11. All the frequency attenuation of lower than 64% of the low cut-off frequency (f_L) and higher than 160% of the high cut-off frequency (f_H) should not be less than 40 dB.

The electronic measuring device should be examined in certain period, during the examination period, the permissible changing range of the examining value can be as $\pm 4\%$.

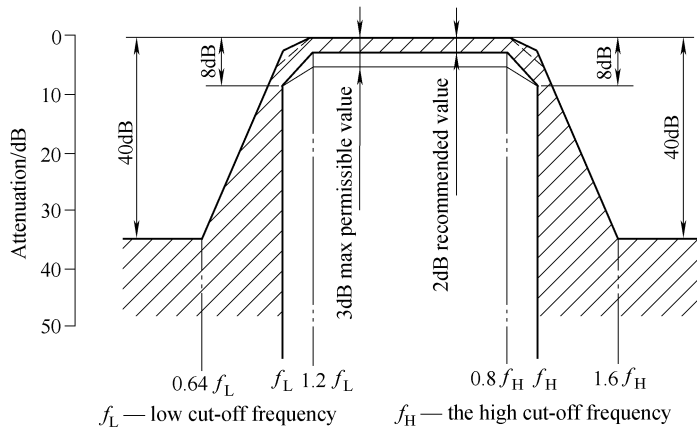


Figure 5-11 Electronic measuring device filtering feature

Inspecting role: The ball vibration velocity are divided as four group: V1, V2, V3, V4. Every group is divided as two frequency bands: low frequency band and high frequency band. Any frequency band vibration velocity of the lot average value and the single ball value should not be over the stipulation value in Table 5 – 40.

In Table 5 – 40:

L_{SW} — single ball low frequency band vibration velocity value;

H_{SW} — single ball high frequency band vibration velocity value;

L_{LW} — ball low frequency band vibration velocity lot average value;

H_{LW} — ball high frequency band vibration velocity lot average value;

Ball vibration velocity lot average value:

Table 5 – 40 Ball vibration velocity limit value

Ball nominal diameter/mm		Ball vibration velocity/ ($\mu\text{m/s}$)							
over	to	V1				V2			
		L_{LW}	L_{SW}	H_{LW}	H_{SW}	L_{LW}	L_{SW}	H_{LW}	H_{SW}
3 ^①	7	22.0	27.5	28.5	35.5	16.5	20.5	22.5	27.5
7	11	30.0	37.5	36.0	45.0	22.5	28.5	27.5	34.5
11	16	40.0	50.0	46.0	57.5	30.0	37.5	35.0	44.0
16	22. 225	61.0	76.0	70.0	87.5	48.0	60.0	56.0	70.0

Ball nominal diameter/mm		Ball vibration velocity/ ($\mu\text{m/s}$)							
over	or	V3				V4			
		L_{LW}	L_{SW}	H_{LW}	H_{SW}	L_{LW}	L_{SW}	H_{LW}	H_{SW}
3 ^①	7	11.0	14.0	16.5	20.5	4.6	6.0	8.9	11.5
7	11	15.0	19.0	19.4	24.0	6.0	7.5	11.0	14.0
11	16	20.0	25.0	24.5	30.5	9.0	11.0	16.0	20.0
16	22. 225	33.0	41.0	39.0	49.0	16.5	20.5	31.0	39.0

① Include 3mm.

For the ball lot that the ball diameter is as 3 ~ 10mm, the measurement should be performed for random sampling of twenty balls per lot and taking

the vibration velocity arithmetic average value of one frequency band as this ball lot vibration velocity average value. For the ball lot that the ball diameter is as 10 ~ 15.5mm, the measurement should be performed for random sampling of ten balls per lot and taking the vibration arithmetic average value of one frequency band as this ball lot vibration average value.

If the ball lot vibration velocity average value of any frequency band is not over the stipulation value in Table 5 - 40, but a single ball maximum vibration velocity value is over, then the exceeded difference value should be added to that lot vibration average value and taking this value as the new judging value. If the value is still not over the stipulation value after adding the exceeded difference value and at the same time, this lot value is also met the stipulation value, then this lot value is judged as qualified.

This standard was submitted by Hangzhou bearing test and research center and was discussed by China rolling bearing standard committee.

4. Deep groove ball bearing vibration peak value standard

(1) Deep groove ball bearing vibration (acceleration) peak value

The bearing vibration acceleration peak value designates the corresponding acceleration degree of the maximum value of the bearing vibration acceleration during the given time interval. Its unit is decibel (dB). The recommended value here is suitable for the foresight judging of the deep groove ball bearing abnormal sound producing probability. The measuring method is according to the deep groove ball bearing vibration (acceleration) measuring method JB/T5314 — 2002. This standard is suitable for the deep groove ball bearings that the diameter is 3 ~ 60mm, the radial clearance is the 0 group, the diameter series is the 0, 2, 3 series and the vibration acceleration degree is the Z3 group or above the Z3 group. And for these bearings, some requirements for peak value has been put forward.

For the deep groove ball bearings that the diameter is 3 ~ 60mm, the radial clearance is the 0 group, the diameter series is the 0, 2, 3 series, the single bearing vibration acceleration peak value should be conformed with the stipulation value in Table 5 - 41.

Table 5-41 Bearing vibration acceleration peak value

(unit: dB)

Nominal bearing diameter/mm	Single bearing vibration acceleration peak value not over					
	diameter series (0)		diameter series (2)		diameter series (3)	
	ZP3	ZP4	ZP3	ZP4	ZP3	ZP4
3	44	39	46	42	47	44
4	44	39	46	42	47	44
5	46	41	48	43	49	45
6	46	41	48	43	49	45
7	47	42	50	44	50	46
8	47	42	50	44	51	46
9	48	43	51	45	52	47
10	49	44	51	46	53	48
12	50	45	51	46	53	48
15	51	46	52	47	54	49
17	51	46	52	47	54	49
20	52	47	54	49	55	50
22	52	47	54	49	55	50
25	54	49	56	52	57	53
28	55	50	57	53	58	54
30	55	50	57	53	58	54
32	56	51	58	54	59	55
35	57	52	59	55	60	56
40	58	53	60	56	61	57
45	61	56	62	59	63	60
50	63	58	64	61	65	62
55	65	60	66	63	67	64
60	67	62	67	64	69	66

The single bearing (for any bearing in the sample) is judged as not qualified if its bearing vibration acceleration peak value is over the stipulation value in Table 5-41.

This standard was approved by China rolling bearing standard

committee (CSBTS 98.80 — 2002).

(2) Deep groove ball bearing vibration (velocity) peak value

In order to promote the development of the motor bearing technical level and to reflect the most bearing user opinion for the abnormal sound of the low noise deep groove ball bearings, the deep groove ball bearing vibration (velocity) peak value standard has been made by Hangzhou bearing test and research center and has been approved to be used in China by China rolling bearing standard committee.

This standard is suitable for the deep groove ball bearings that the diameter is 3 ~ 60mm, the radial clearance is the 0 group, the diameter series is the 0, 2, 3 series. Usually, the low noise requirement is just put forward for the vibration V3, V4 group or over the V3, V4 group deep groove ball bearings. Therefore, this standard stipulated the P3, P4 groups peak limit values for the deep groove ball bearings of the 0, 2, 3 diameter series, the V3, V4 vibration groups. This standard can be used in the manufacturer production inspection and the user acceptance.

The bearing vibration (velocity) measuring method is according to the JB/T5313 — 2000 (this chapter 5.2). The bearing vibration (velocity) peak limit values are shown in Table 5 – 42.

Table 5 – 42 Bearing vibration (velocity) peak limit value

(unit: $\mu\text{m/s}$)

Nominal bearing diameter d/mm	P3			P4		
	low frequency band	middle frequency band	high frequency band	low frequency band	middle frequency band	high frequency band
3	95	80	75	85	50	50
4	95	80	75	85	50	50
5	105	105	90	95	55	55
6	105	105	90	95	55	55

(continued)

Nominal bearing diameter d /mm	P3			P4		
	low frequency band	middle frequency band	high frequency band	low frequency band	middle frequency band	high frequency band
7	130	140	120	115	60	60
8	130	140	120	115	60	60
9	130	140	120	115	60	60
10	165	175	150	135	70	75
12	165	175	150	135	70	75
15	195	230	175	155	90	90
17	195	230	175	155	125	125
20	240	230	225	180	125	125
22	240	300	225	180	150	160
25	240	300	225	180	150	160
28	240	300	225	180	175	200
30	270	375	300	210	175	200
32	270	375	300	210	175	200
35	270	375	300	210	210	225
40	330	450	400	245	250	225
45	330	450	400	245	300	300
50	375	500	500	285	350	350
55	375	550	550	285	350	350
60	435	550	650	300	400	400

5. Other rolling bearing standard requirement for bearing vibration

Except the above bearing vibration standards, there are also related stipulations for bearing vibration level in other rolling bearing standards.

(1) Rolling bearing production quality degree dividing standard CSBTS

TC98.77 — 2001

This standard has stipulated that for the rolling bearings that will be participated the first degree production evaluation or the excellent production evaluation, the bearing vibration has to be measured. For the deep groove ball bearings that the bore diameter is $\leq 60\text{mm}$, the vibration velocity (or acceleration) must be reaching or above the V3 group level (or Z3 level). For the deep groove ball bearings that the bore diameter is $> 60\text{mm}$, the vibration velocity (or acceleration) must be reaching or above the V2 group level (or Z2 level). This standard has also stipulated the evaluation rules for the excellent production, the first degree production and the qualified production. For the tapered roller bearings and the cylindrical roller bearings, this standard has also stipulated the evaluation rules for the excellent production, the first degree production and the qualified production.

(2) Precise rolling bearing technical prerequisite CSBTS TC98.76 — 2001

This standard has stipulated that for the precise deep groove ball bearings, their vibration velocity (or acceleration) limit value must be reaching the V3 (or Z3) group value; for the precise cylindrical roller bearings, their vibration velocity (or acceleration) limit value must be reaching the V2 (or Z2) group value; for the precise tapered roller bearings, their vibration velocity (or acceleration) limit value must be reaching the V2 (or Z3) group value.

5.3 Rolling Bearing Noise and Measurement

5.3.1 Relationship of Rolling Bearing Noise with the Bearing Vibration

While the rolling bearing vibration spreads to the radiation surface, the vibration energy will be transformed into the compressive wave that will be radiated out through the air medium, in which the part of $20 \sim 20000\text{Hz}$ will be heard by the people ear, that is the rolling bearing noise. Therefore, the rolling bearing vibration is the source of the noise wave. The different bearing noise source comes from the different type of bearing vibration.

Table 5 – 43 listed the relationship of the rolling bearing noise with the bearing vibration.

Table 5 – 43 Relationship of the rolling bearing noise with the bearing vibration

Vibration type	Produced noise	Noise feature	Improve method
rolling bearing structure vibration	railway sound crushing sound (in cylindrical roller bearing)	① this sound is the rolling bearing characteristic sound, common bearing sound, that is the basic sound in the bearing sounds if the railway sound adding other sound ② the frequency is over 1000Hz ③ the main noise frequency is not changed with the rotation speed, but the sound pressure degree increases with the rotation speed	① improving the structure design ② raising manufacturing precision ③ keep good using condition
rolling bearing component raceway surface waviness causing vibration	high frequency vibration sound	① the sound frequency is changed with the rotation speed ② the main reason causing bearing noise is the waviness of the component surface	improving the bearing component surface manufacturing quality, reducing the waviness amplitude
cage vibration	cage sound	① cage sound is the cage vibration causing sound by the cage colliding with the rolling element ② this sound periodically occurs	① reducing ball bearing radial clearance or preloading ② reducing bearing mounting error ③ using good lubricant ④ raising cage making precision
dust causing vibration	dust sound	the non-periodically sound caused by the dust particle and the impurity is called as the dust sound, this sound degree is not certain	① cleaning out the impurity in the lubricant ② improving bearing cleaning method and the sealing type

5.3.2 Rolling Bearing Noise Expressing Method

In rolling bearing noise measuring, the using physical quantities are shown in Table 5 – 44 and Table 5 – 45.^①

Table 5 – 44 Expressing bearing noise physical quantities (1)

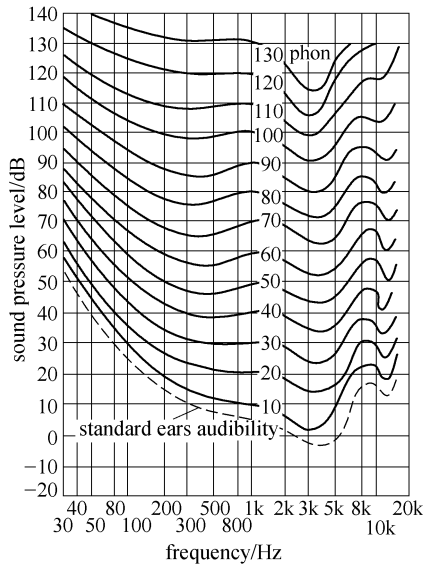
physical quality	sound pressure	sound intensity	sound power	degree of loudness
definition	the sound pressure is the varying amount of the atmosphere pressure intensity increase or decrease caused by the sound wave, expressed by p	the sound intensity is the sound energy average value that has been through per area, per time in the direction perpendicular to the sound wave spreading direction, expressed by I	the sound power is the total sound energy radinted per time. expressed by W	the degree of loudness is the comprehensive evaluation of the sound degree for strong or weak sound that were receipted by the people ears
unit	N/m^2	W/m^2	W	sone
note	the sound pressure can be positive or negative, usually using the sound pressure effective value	relationship of the sound pressure with the sound intensity $I = \frac{p}{\rho c}$	relationship of the sound power with the sound intensity $W = \int_s I dS$	

note: ρ — medium mass (kg/m^3); c — sound speed (m/s); S — sound source including area (m^2).

① The physical quantity in Table 5 – 44 is expressing the absolute value of the sound strength degree or weakness degree, but it is not convenient. At present, the commonly used physical quantity for expressing the sound strength degree or weakness degree is in Table 5 – 45.

Table 5-45 Expressing bearing noise physical quantities (2)

Physical quality	Sound pressure level	Sound intensity level	Sound power level	Degree of loudness level
expression	$L_p = 20 \lg \frac{p}{p_0}$	$L_I = 20 \lg \frac{I}{I_0}$	$L_W = 10 \lg \frac{W}{W_0}$	taking the 1000Hz pure sound as the basic sound in the free sound field. If some sound heard as the same loudness as the basic sound, this sound level of the degree of loudness is equal to the basic pure sound pressure level, the unit is expressed by phon. The curve of the equal degree of loudness dB is shown in Figure 5-12.
basic value	$p_0 = 2 \times 10^{-5} \text{ N/m}^2$	$I_0 = \frac{p_0^2}{\rho c}$	$W_0 = 10^{-12} \text{ W}$	
unit	dB	dB	dB	

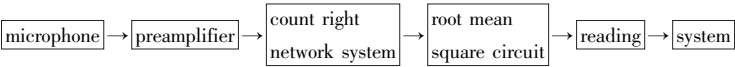
**Figure 5-12 Curve of equal degree of loudness**

5.3.3 Rolling Bearing Noise Measurement

The rolling bearing noise measurement should be performed in the special anechoic chamber. The basic noise in the anechoic chamber is very low, therefore the bearing noise can be separated with the surrounding

noise. But the building cost of anechoic chamber is very high and the measurement on-the-spot can not be performed. Therefore, it is the general to use the bearing vibration measurement instead of the bearing noise measurement in the anechoic chamber.

The basic and comprehensive using instrument in the noise measurement is the sound pressure degree meter, its principle is as:



The microphone transforms the sound pressure wave into the electric signal wave. The count right network system^① modifies the noise frequency resonance feature and modifies it as A, B, C count right signals or the linear count right signals according to the ISO standard. Figure 5 – 13 shows the frequency resonance of A, B, C count right signals.

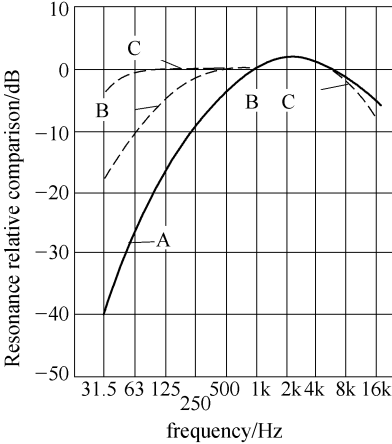


Figure 5 – 13 A, B, C count right network frequency resonance

① In order to let the measuring instrument approximately conformed with the people ear resonance for the noise sound, it is that simulating the people ear hearing sensation on the sound measuring instrument to diminish or to increase in different scale for the different sound signal. The read sound degree level is called as the count right network sound degree level. In recent years, a count right is normally used, it is called as A sound degree level, dB (A). The A count right is simulating the 40 phon pure sound resonance for the people ear. The B count right is simulating the 70 phon pure sound resonance for the people ear. The C count right is simulating the 100 phon pure sound resonance for the people ear.

5.4 The International Standard for Rolling Bearing Vibration Measurement

International Standard organization ISO Fourth Committee ISO /TC 4 has endeavored to make the rolling bearing vibration measuring method standard for many years. There are three parts in ISO 15242 — Rolling bearing vibration measuring method:

first part: Fundamentals

second part: Radial ball bearings with cylindrical bore and outside diameter surface

third part: Aligning roller bearing and tapered roller bearing with cylindrical bore and outside diameter surface

In the year 2000, ISO published the first Rolling bearing vibration measuring method draft: ISO CD15242 — 1:2000 rolling bearing- vibration measuring method-first part: fundamentals for asking the opinion in the world range. On the May first of 2004 year, through the approval by the international organization members voting, ISO published officially the first Rolling bearing vibration measuring method standard: ISO 15242 — 1 Rolling bearing vibration measuring method first part: fundamentals and then on the Oct. first of 2004 year, ISO published officially the first Rolling bearing vibration measuring method standard: ISO 15242 — 2 Rolling bearing vibration measuring method second part: radial ball bearing with cylindrical bore and outside diameter surface.

5.4.1 Part 1: Fundamentals

This part of the standard mainly stipulated the general principle in bearing vibration measuring, such as the term definition, the basic concepts, the bearing measuring procedure, the measurement and evaluation method, the measured bearing condition, the measuring surrounding condition, the calibration and the measuring system evaluation, the contacting resonance and so on. The detailed contents will be discussed as following.

1. Term and definition

The following terms are suitable for this standard

(1) Error motion

The rotation axial line radial, axial (translational) motion and tilting (angle) motion that are not the expecting motion, but not included the motion caused by temperature varying and load varying.

(2) Stiffness

The ratio of an elastic component force (or moment) varying value with the varying value of related translational motion (rotation) displacement.

(3) Vibration

Variable which is describing a machinery system motion or location parameter that is a varying value with the time that can be larger or smaller than one average value or a reference value.

(4) Transducer

A device that can receive a system energy and can also transport the same type energy or the different type energy to another system, and can display out on the export end.

(5) Electromechanical pickup

A kind of transducer which can be stimulated by a machinery system energy, (such as stress, force, motion energy) and can supply to the electrical system, or on the contrary.

Note: Within the vibration measuring and shocking measuring, the main transducer types are as:

- 1) piezoelectric acceleration meter.
- 2) piezoresistive acceleration meter.
- 3) Strain acceleration meter.
- 4) Variable resistance pick up.
- 5) Static electricity (capacity) pick up.
- 6) Coupling (chaff) strain pick up.
- 7) Variable magnetic resistance pick up.
- 8) Magnetic expand and contract pick up.
- 9) Moving conductor pick up.

10) Moving coil type pick up.

11) Induction pick up.

(6) Displacement

A vector that is describing a body or a particle location varying relative to the reference system.

(7) Velocity

A vector that is describing the displacement differential with time.

(8) Acceleration

A vector that is describing the velocity differential with time.

(9) Filter, wave filter

A device that can divide the vibration according to the vibration frequency, and can diminish the wave vibration in relative smaller scale within one frequency or more frequencies, but also can diminish the wave vibration in relative larger scale on other frequencies.

(10) Band-pass filter

A kind of filter that the single vibration band can be transmitted through from the lower cut off frequency larger than zero to the upper cut off frequency.

(11) Pass-band

The frequency between the upper cut off frequency and the lower cut off frequency (band-pass filter frequency).

(12) Nominal upper and lower cut-off frequencies, cut-off frequency)

Higher than or lower than a filter maximum resonance frequency; on these frequencies, the resonance for the sine signal lower than the maximum resonance should be 3 dB.

(13) $V_{r.m.s.}(t)$ (root mean square velocity)

The velocity value that has been got within t time interval taking the mean value after adding the velocity square values for every time intervals, and then taking the extraction of a root.

Note: The root mean square value also can be used for displacement and acceleration.

(14) Exponential mean effective velocity (e. m. e) $V_{e.m.e.}(t)$

A kind of parameter to get the mean velocity, that is resembled to the root mean square velocity, but has considered the exponential diminishing.

Note: 1) The exponential mean effective value also can be used for displacement and acceleration.

2) The exponential mean effective value also can be called as the exponential mean value, or the time diminishing value.

(15) Period

The independent variable minimum increment of a cycle parameter, the function is itself cycling.

2. Basic concepts

(1) Rolling bearing measurement

Figure 5 – 14 shows the basic elements of the rolling bearing vibration measurement and every influencing factors for the vibration measurement.

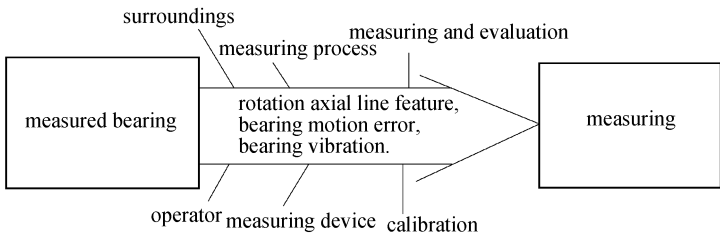


Figure 5 – 14 Bearing vibration measuring basic elements

(2) Rotation axial line feature

The rotating rolling bearing should be supplying a rotating line while a machinery element rotating relative to another machinery element and can carry the radial load and or the axial load. Its rotation axial line could have six direction freedoms. Figure 5 – 15 shows the six direction freedom sketch as follows.

1) Rotating motion, as shown in Figure 5 – 15b.

2) Translational motion in the radial direction, that is the translational motion that is in one or two crossing planes through the rotation axial line, as shown in Figure 5 – 15c and d.

3) Translational motion in an axial line direction, that is the translational motion that is paralleling to a rotation axial line direction as shown in Figure 5 – 15e.

4) Tilting motion in an angle direction, that is the angle direction

motion that is in one or two crossing planes through the rotation axial line, as shown in Figure 5 – 15f and g.

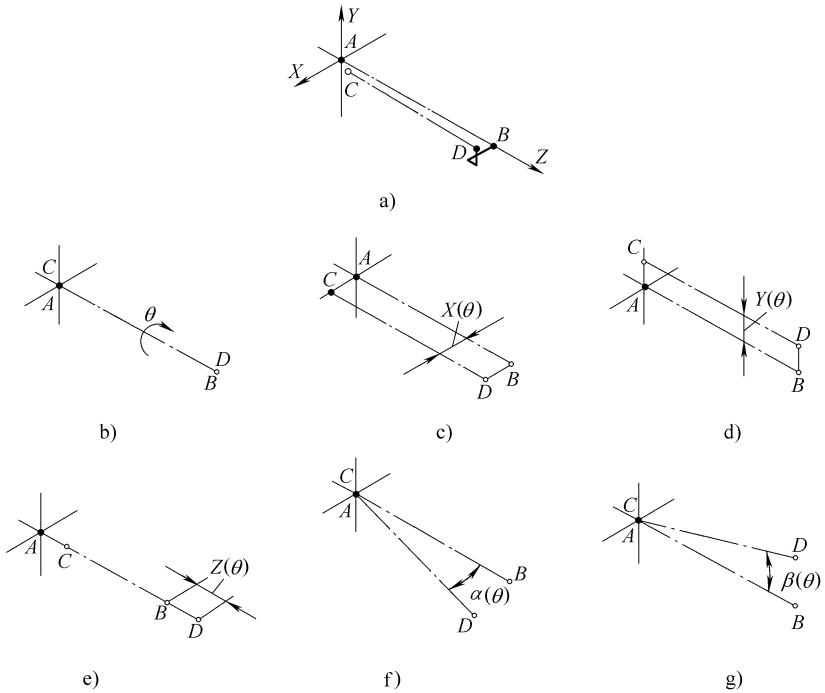


Figure 5 – 15 Rotation axial line six direction freedom sketch

- a) general condition indicating the coordinate direction
 - b) co-axis rotating motion with the Z axis of the reference system
 - c) the radial direction translation motion in the X direction
 - d) the radial direction translation motion in the Y direction
 - e) the axial direction translation motion in the Z direction
- $AB = Z$ reference system axis $CD =$ rotating axis
- f) the tilting motion relative to the original point in the X direction
 - g) the tilting motion relative to the original point in the Y direction

For a running rolling bearing, idealized to say, there is no resistance force for the outside force, that is to say that the friction moment is zero. According to the outside load type, it should be designed as that the rolling bearing not only can support the outside load, but also can have the stiffness

in any freedom or all of the remaining five freedoms. For example, a rolling bearing with the aligning feature can carry both the radial load and axial load, but idealized to say, there should be no stiffness in two tilting directions. Besides, other rolling bearings could be designed as that the free axial motion is permissible, but at the same time, there are stiffness both in the radial direction and in the tilting direction.

(3) Bearing motion error

The rolling bearing should be designed as that the load can be applied in any non-rotating freedom by the bearing. There are 5 non-rotating freedoms. Therefore, the definition of the bearing motion error should be as within the 5 non-rotating freedoms, the bearing rotating axis displacement on any non-rotating freedom direction. Any bearing displacement related to bearing rotating is included, but the displacement caused by temperature change or by outside load change is not included. The bearing motion error is expressed by the displacement, but it also expresses the deviation with the ideal rotating axis. The bearing motion error is caused by the relative motion inside every surface geometry defect while the bearing is rotating. Some of the geometry defects could be the bearing inherent feature (such as the manufactured surface geometric error in manufacturing) but it also could be the bearing component defect produced in mounting or installing

(4) Rolling bearing vibration

The factors to cause bearing motion error also can cause bearing component dynamic vibration. The vibration is the result to cause the displacement change by the motion error. But from another aspect, the inertia force acting related to acceleration and the bearing installation stiffness feature also could cause the inside force. The bearing component deformation changes with time, some non-expecting rolling element and cage motion form and the cage periodic displacement relative to rolling element or ring also can cause the inside force. Vibration is produced by the motion error under special surrounding condition, such as rotation speed and load etc. Bearing vibration can influence the machinery system feature, and the air noise can be produced by the machinery system including the bearing.

3. Bearing vibration measuring process

(1) measuring base

The purpose of ISO 15242 first part is to evaluate the structure vibration for a rotating rolling bearing measured by the transducer. The transducer type can be the velocity type, the acceleration type or the force type. The transducer is installed on the special point of the rolling bearing, or on a machine component special point of a tester that is machinery coupling with a bearing ring. The transducer acting line (that is radial or axial) must be consistent with the relative reference system speculating position. Under the stipulating condition, the bearing is rotating in a fixed rotating speed, and the transducer signal in a stipulating period is monitored. Then, analyzing the the obtained data, one or more parameters can be calculated out for expressing the bearing vibration level. From these measuring result, the bearing vibration data can be got under the selected test condition. Under different running condition, the conclusion about the relation of the bearing vibration with the noise can not be got from these measuring results.

Figure 5 - 14 shows the bearing vibration measuring process sketch.

(2) Rotating speed

The bearing vibration measurement is performed on the bearing dynamic condition. While measuring, the bearing outer ring is in stationary or in gradually rotating, the bearing inner ring is rotating in fixed rotating speed according to the bearing dimension value and the bearing structure.

During measuring, the actual rotation speed can not be over 1% of the stipulated speed, but can not be lower than 2% of the stipulated speed.

(3) Bearing rotating axis direction

The bearing vibration measurement can be performed in the vertical running shaft position, or in the horizontal running shaft position. When measuring in the horizontal running shaft position, the earth gravitational force direction change relative to the running rolling element should also be considered. Unless the rolling element centrifugal or the contact force on the rolling element is larger than the itself gravity, these factors also could cause

additional vibration.

(4) Bearing load

For reaching the limited running condition within the bearing, in bearing vibration measuring, the load should be applied on the bearing. The load should be large enough for preventing the spin sliding produced between the rolling element and the raceways of the inner ring and the outer ring, so that the measuring result should not be disturbed.

(5) Transducer

The measuring quantity should be the bearing outer ring radial vibration value or the bearing outer ring axial vibration value. An electromechanical pickup can transform the machinery motion into an electrical signal. The given signal should be proportional with the displacement, velocity or acceleration. The force pickup also can be used, but it has to transform the signal into one parameter of the above three parameters.

It should be very clear that the measuring system is a contact type measuring system or a non-contact type measuring system. The non-contact type measuring system is specially suitable for the displacement measuring. For the contact type measuring system, it must be care that the contact type measuring system transducer should be contacted with the producing vibration outer ring, and it should be also care to guarantee that the bearing outer ring vibration must not be influenced. In order to catch all the vibration signals in the proper frequency range, on the contrary, there should be enough stability for the contacting. In order to get this purpose, the motion mass should be as small as possible. If the vibration is transformed through the transducer tip that is contacting with the bearing outer ring, the resonance vibration should be considered. Details will be introduced in Section 4. 1. 7.

The bearing outer ring vibration motion is a process of complicately superimposing varied amplitude displacement at different frequency. There also is the possibility for existing a single higher amplitude, especially in the high frequency condition (especially in the case of bearing with defects). Usually to say, the amplitude decreases with the increase of the frequency.

In the condition of several thousand Hz range, it perhaps can be decreased to the nanometer level. Therefore, it is very difficult to give reliable measuring results in high frequency range by the displacement transducer system. From another aspect, the acceleration transducer system is very suitable for high frequency vibration measuring, but for low frequency range vibration measuring, the high dynamic performance is needed for the vibration measuring. The velocity type transducer system is the better measuring system. Within the velocity type transducer system, the signal is expressed in proportional to the velocity. In the case of a need, the gathered signal can be transformed as the electrical signal by the known calibrated transforming factor.

4. Bearing vibration measuring and evaluation method

(1) Measuring physical quantity

The measuring physical quantity is the velocity, expressed by $V_{r.m.s}$ ($\mu\text{m/s}$). The measuring direction can be the radial direction or the axial direction that is determined by the measured bearing type.

(2) Frequency domain

The measured signal should be within 50 ~ 10000Hz, the velocity signal can be measured within one frequency band or within many frequency bands. It is suggested that for different bearing types, the special frequency range should be adopted.

Note: For example for radial and angular contact bearing within certain dimension range, 50 ~ 300Hz, 300 ~ 1800Hz, 1800 ~ 10000Hz frequency range can be used.

The frequency analyzing method can be used for analyzing vibration signal.

(3) Time domain

The measured peak value or the maximum pulse value of the velocity signals within the time domain range can be used as a kind of supplementary selection to achieve the agreement between the manufacturer and the user. Generally, the higher measured vibration peak value or the maximum pulse value can be easily caused by the bearing component surface with defect or the measured bearing being polluted. There could be varied different evaluation methods according to bearing types and applications.

(4) Transducer frequency resonance and filter feature

The electromechanical pickup resonance should be in the stipulated range in Figure 5 – 16.

In Figure 5 – 16, the lowest transducer frequency resonance requirement should include the amplifier compensation export signal.

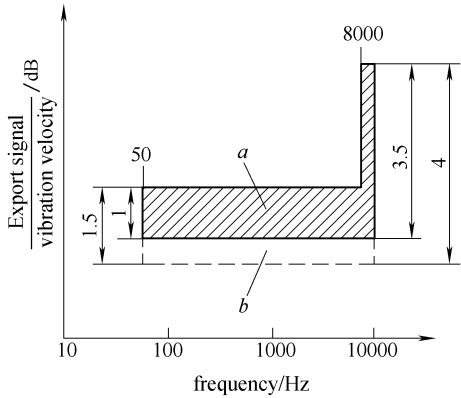


Figure 5 – 16 Transducer frequency resonance stipulation value

a) recommended range region b) maximum permissible region (include a)

Amplifier linearity: The maximum linear deviation within 10 μ m/s and 3000 μ m/s range should be less than 10% of the bearing vibration amplitude.

Transducer sensitivity: The transducer sensitivity matched with the electronic instrument should be stipulated within $\pm 5\%$. In the transducer static axial displacement range, this transducer sensitivity should be in the stipulation range. The proper compensation should be supplied by the electronic instrument in the case that there is some changes in the sensitivities between every transducer.

The electronic instrument filter feature should be in the stipulated band-pass limit range in Figure 5 – 17. For all the frequencies of lower than 64% low cut-off frequency (f_{LOW}) and all the frequencies of higher than 160% high cut-off frequency (f_{high}), the band-pass attenuation should be not lower than 40 dB.

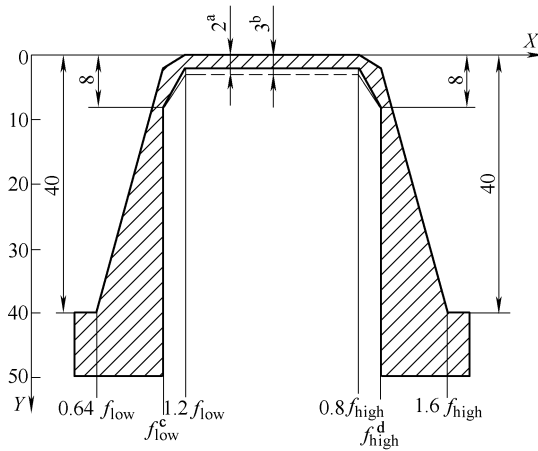


Figure 5-17 Filter feature

X — cutoff frequency Y — attenuation value a — recommended range

b — maximum permissible range c — nominal low cutoff frequency

d — nominal high cutoff frequency

(5) Taking average time method

During the velocity measuring in every frequency band range, the average of the representative time readings should be read within not less than 5s when the vibration signal achieves the stable level. So called achieving the stable level designates that there is only occasional random waving in the average value nearby. As for which average time equation will be selected, the manufacturer and the user should determine by discussion. The root mean square velocity and the exponential mean effective velocity are two typical taking average time equations. Their definitions have been given in Section 4. 1. 1. Their mathematical expressions are as follows:

The root mean square velocity value:

$$V_{r. m. s.}(t) = \sqrt{\left(\frac{1}{T}\right) \int_{(t=T)}^t V^2(t') dt'} \quad (5-1)$$

In the equation: $V(t')$ — vibration velocity change with time;

T — sampling time. The sampling time should be longer than the period of any main frequency component of $V(t')$.

The root mean square velocity $V_{r.m.s}$ should be always taken at $t = T$.

The exponential mean effective velocity value:

$$V_{e.m.e.}(t) = \sqrt{\frac{1}{T} \int_0^t V^2(t') e^{-(t-t')/\tau} dt'} \quad (5-2)$$

In the equation: $V(t')$ — vibration velocity change with time;

τ — attenuation period, which should be longer than the period of any main frequency component of $V(t')$.

The exponential mean effective velocity value $V_{e.m.e.}(t)$ should be always read in $t > \tau$ time interval.

The stable condition should be achieved within 5 mins from the beginning evaluation. If the stable condition can not be achieved within 5 mins, a proper measuring time should be determined by discussing jointly with the manufacturer and the user.

(6) Measuring order

Measuring should be performed on several required positions. For special bearing types, it will be discussed in details in other part of ISO 15242. (for radial ball bearing, it will be discussed in details in ISO 15242-2).

5. Measuring conditions

(1) measured bearing conditions

1) Lubrication in advance

For the bearing that the lubricant (lubricating grease, lubricating oil or solid lubricant) has been filled in advance, including the bearings with seals and with shields, the measuring should be performed on the original condition.

For the non-lubricated bearing in advance, measuring should be performed according to the following (2) and (3) processes.

2) Bearing cleanness. Because the dirty and the impurity could influence the bearing vibration level, it should be the first step to wash the measured bearing carefully. It should take special care that the dirty and the impurity or other things caused vibration can not be included into the measured bearing.

Note: Some anti-rusting agents smearing over the bearing can satisfy the lubrication requirement for bearing vibration measurement. In this case, it is not required to re-clean for removing the anti-rusting agent.

3) Lubrication. Before measuring, the measured bearing should be lubricated by the low viscosity lubrication oil according to the bearing type and the dimensions. In order to let the lubricant distributing evenly in the bearing, it is the right way to rotate the bearing.

(2) Testing environment condition

The bearing measuring should be performed at the room temperature. The bearing vibration should not be influenced by environment condition.

(3) Requirement for the tester

1) Spindle/mandrel supporting stiffness

The spindle (including the mandrel) is used for supporting the bearing and driving the bearing inner ring rotating. Besides, excepting for transforming the rotating motion, the spindle (including the mandrel) also is the rigid reference system for the inner ring axis. The vibration transformation between the spindle/mandrel support and the bearing inner ring can be omitted within the used frequency band range, comparing with the measured vibration velocity. (If there is a controversy, the precise value can be taken by discussing between the manufacturer and the user).

2) Loading mechanism

For the loading mechanism used for loading the bearing outer ring, it is ideal that the bearing vibration is not influenced in all of the radial direction, the axial direction, the angle direction and the bending direction (determining based on the bearing type) and the outer ring is basically in the free vibration condition.

3) Requirement for the load value and the centralization

For the special bearing types, it will be discussed in details in other part of ISO 15242. For the radial ball bearing, Table 5 – 47 and Table 5 – 49 show these values.

4) Transducer axial position and measuring direction

For the special bearing types, these will be stipulated in details in other part of ISO 15242. For the radial ball bearing, these are shown in

Figure 5 – 18 and Figure 5 – 19.

5) Mandrel

For the mandrel cylindrical surface for mounting bearing inner ring, its outer diameter should be manufactured according to the f5 grade tolerance of ISO 286-2 and the geometric deviation should be the minimum value, in order to guarantee mounting the mandrel in slide fit into the bearing bore.

(4) Requirement for the operator

The qualified operator should guarantee to conduct the bearing vibration measuring according to ISO 15242 first part and other part relative to bearing type.

6. Measuring system calibration and evaluation

(1) Generals

In order to guarantee that the measuring system should be calibrated in time before bearing measuring, the calibration should be performed according to the stipulated process.

(2) Measuring system every unit calibration

Every unit in the bearing vibration measuring system should be calibrated as follows:

- 1) The driving mechanism makes bearing rotating.
- 2) The loading mechanism applies load to the bearing.
- 3) The transducer system transforming the bearing vibration into the electrical signal
- 4) Electronic instrument system device deals with electrical signal (amplifier, filter, indicator).

Every part of the measuring system must keep their original designing performance and can be adjusted under the controlled condition. The adjustment or the calibration should be traced back to the relative stipulation in the international measuring standard or other country standards. The following every part is the main calibration item and the confirmation item:

- 1) The driving mechanism :
 - ① The spindle rotation speed.
 - ② The main motion error and the remaining vibration.

③ The mounting bearing spindle, mandrel condition (damaging, rusting, deformation and dimension change etc.).

2) The loading mechanism:

① The load value.

② The loading direction and the centralization degree with the spindle axis.

③ The loading point position.

3) Transducer:

① The sensitivity and the amplitude linear degree.

② The frequency resonance.

③ The direction and the position.

4) Electronic instrument system device (amplifier, filter, indicator):

① The amplifying times and the linearity.

② The frequency feature.

③ The instrument or data indicating precision.

(3) System performance evaluation

If finishing the measuring using a bearing component under the same position and by the same measuring device and the same tester parameter, the measuring repeatability must be within 10% of the measuring value.

Note: The measuring system change does not include the measured bearing change.

7. Contacting resonance

In international standard ISO 15242-1, there is a special appendix A as a reference data for explaining the contacting resonance.

(1) The contacting force

If the transducer is loaded by the spring, in order to prevent the transducer away from contacting with the outer ring, the applied contacting force should be larger than ma (here m is the movable part mass of the transducer, but a is the measured maximum acceleration).

(2) The contacting resonance

The transducer measuring tip has its own material modulus of elasticity. Therefore, the transducer measuring tip can play the role as a spring, so there is existing the contacting resonance. If the contacting tip is a spherical shape, the situation becomes more complicated, because the

measuring tip plays the role of a spring at this time, and its rigidity is changeable and is increasing with the load. The higher the material modulus of elasticity E is and the larger the radius of the contacting tip is, the higher the resonance frequency is. Table 5 – 46 has given some examples that the transducer contacting tips are the half-spherical shapes: the half-spherical shape transducer contacting tip material modulus of elasticity is $E = 600$ GPa; the transducer overall motion mass is m ; pressing the motion mass onto the outer ring outside surface by the static force F . The bearing outer ring material modulus of elasticity is $E = 200$ GPa.

Table 5 – 46 Contacting resonance

r/mm	F/N	m/g	f/kHz
1	1	1	9.6
5	1	1	12.6
1	5	1	12.6
1	1	5	4.3

5. 4. 2 Part 2: Radial Ball Bearings with Cylindrical Bore and Outside Surface

This standard stipulates the single row and double row radial ball bearing vibration measuring method. This standard is suitable for the single row and double row radial ball bearings that the contact angle is smaller than 45° , and the bore surface and the outer diameter surface are the cylindrical surface, but this standard is not suitable for the filling slot radial ball bearings and the three-point or four-point contact radial ball bearings. The terms have been explained in Section 5. 4. 1.

1. Bearing measuring process

(1) Rotation speed

The rotation speed should be 30s^{-1} ($1800\text{r}/\text{min}$), the permissible error is $1\ (1/2)\%$.

Other rotation speeds can be used if there is an agreement between the manufacturer and the user. In order to get the sufficient vibration signal, for

the smaller dimension range bearings, higher rotation speed can be selected for use [$40 \sim 60\text{s}^{-1}$ ($2400 \sim 3600\text{r/min}$)]. On the contrary, in order to prevent the ball and the raceway damage, for the larger dimension range bearings, lower rotation speed can be selected for use [$10 \sim 20\text{s}^{-1}$ ($600 \sim 1200\text{r/min}$)].

(2) Bearing axial load

The acting load on the bearing should be the axial load, and the load value is listed in Table 5 – 47.

Other load values can be used if there is an agreement between the manufacturer and the user. That is to say that in order to prevent the sliding between the ball and the raceway, the axial load higher than the listed axial load can be selected to use according to the bearing design and the used lubrication condition. Or in order to prevent the ball and the raceway damage, the axial load lower than the listed axial load can be selected to use.

Table 5 – 47 Bearing axial load value

Bearing outer diameter		Single row and double row deep groove and aligning ball bearing		Single row and double row angular contact radial ball bearing			
>	≤	min	max	contact angle > 10° ≤ 23°		contact angle > 23° ≤ 45°	
mm		N		N			
10	25	18	22	27	33	36	44
25	50	63	77	90	110	126	154
50	100	135	165	203	247	270	330
100	140	360	440	540	660	720	880
140	170	585	715	878	1072	1170	1430
170	200	810	990	1215	1485	1620	1980

2. Bearing vibration measuring and evaluation

(1) Measuring physical quantity

The measuring physical quantity is the radial vibration velocity $V_{r, m, s}$ ($\mu\text{m/s}$).

(2) Frequency range

The measuring frequency range should be within one or more frequency band ranges in Table 5 – 48.

Table 5 – 48 Frequency range

rotation speed ^①		low frequency band (L)		middle frequency band (M)		high frequency band (H)	
min	max	low	high	low	high	low	high
r/min		Hz		Hz		Hz	
1764	1818	50	300	300	1800	1800	10000

Note: Other frequency range that there is an important meaning for the bearing smoothly running can be selected for use if there is an agreement between the manufacturer and the user.

- ① For the case that the rotation speed is different than the nominal rotation speed 1800r/min, in order to close to the actual reality, it should be adjusted according to the rotation speed ratio. For the frequencies lower than 50Hz or higher than 10000Hz, unless there is an agreement between the manufacturer and the user, usually, it can not be used.

(3) Peak value measurement

If there is an agreement between the manufacturer and the user, it can be considered as a supplement selection to detect the peak value or the maximum pulse value. In general condition, larger peak value or maximum pulse value could be existing in the case that there is defect on the bearing surface, or the measured bearing is polluted. There are many kinds of evaluation methods for selection according to the bearing type and the application condition.

(4) Measuring order

Except for the single angle contact ball bearing, the measurement is performed under the axial load acting at one side of the outer ring for all type bearings and then repeated once under the axial load acting at another side of the outer ring. For the single angle contact ball bearing, the measuring can be only carrying on at one outer ring side that the axial load is acting.

In order to diagnosing purpose, the measurement can be performed for many times at different angle positions relative to the transducer.

For the bearing acceptance, the maximum reading in corresponding frequency band should be within the limit range determined by the manufacturer and the user.

The measuring time is: The reading is read within the period not less than 5 s after the velocity signal has achieved the stable level within every frequency band. So called achieving the stable level designates that there is only occasional random waving in the mean value nearby.

3. Measuring conditions

(1) measured bearing condition

1) Lubrication in advance

For the bearing that the lubricant (lubricating grease, lubricating oil or solid lubricant) has been filled in advance, including the bearings with seals and with shields, the measurement should be performed on the original condition.

Note: For the non-lubricated bearing in advance, the measurement should be performed on according to the following (2) and (3) processes. The measured vibration level could be increasing or decreasing.

The following items 2), 3) are usually suitable for the non-lubricated bearings in advance. But that could be used for seeking the non-accepted reason while there is some controversies for accepting the vibration level.

2) Bearing cleanness. Because the dirty and the impurity could influence the bearing vibration level, it should be the first step to wash the measured bearing carefully. It should take special care that the dirty and the impurity or other things caused vibration can not be included into the measured bearing.

Note: Some anti-rusting agent smearing over in the bearing can satisfy the lubrication requirement for bearing vibration measurement. In this case, it is not required to re-clean.

3) Lubrication. Before measuring, the measured bearing should be fully lubricated by the filtered oil (by $0.8\mu\text{m}$ maximum filter). The lubrication oil viscosity should be within $10 \sim 100\text{mm}^2/\text{s}$.

During lubrication, the bearing should be slowly rotated to evenly distribute the lubrication oil.

Note: In order to suit for the bearing application requirement, if there is an agreement

between the manufacturer and the user, other viscosity lubricants can be used.

(2) Testing environment condition

The bearing measurement should be performed under the room temperature. The bearing vibration should not be influenced by environment conditions.

(3) Requirement for the tester

1) Spindle/mandrel supporting stiffness

The spindle (including the mandrel) is used for supporting the bearing and driving the bearing inner ring rotating. Besides, excepting for transforming the rotating motion, the spindle (including the mandrel) is the rigid reference system for the inner ring axis. The vibration transformation between the spindle/mandrel support and the bearing inner ring can be omitted within the used frequency band range, comparing with the measured vibration velocity. (If there is a controversy, the precise value can be taken by discussing between the manufacturer and the user).

2) Loading mechanism

For the loading mechanism used for loading the bearing outer ring, it is ideal that the bearing vibration is not influenced in all of the radial direction, the axial direction, the angle direction and the bending direction (determining based on the bearing type) and the outer ring is basically in the free vibration condition.

3) Requirement for the outside acting load value and the centralization

For the radial ball bearing, the outside acting stable axial load should be acted on the outer ring according to the stipulation value in Table 5-47.

The bearing tilting caused by contacting with the machinery element can be omitted comparing with the bearing inherent geometric precision.

The outside acting axial load position and the direction should be coincided with the spindle axis, the error should be within the stipulation value in Table 5-49.

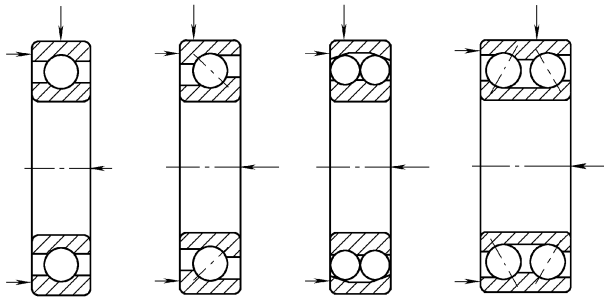
4) Transducer axial position and measuring direction

For the radial ball bearings, the transducer axial position and measuring direction are as follows:

Table 5 – 49 Loading axis deviation related to bearing inner ring rotating axis

Bearing outer ring diameter D		Radial direction deviation with bearing rotating axis $H_{\max}$	Angle direction deviation with bearing rotating axis
$>$	$\leq$		
mm		mm	($^{\circ}$)
10	25	0.2	0.5
25	50	0.4	
50	100	0.8	
100	140	1.6	
140	170	2	
170	200	2.5	

Correct axial position: on the outer ring outside surface, its position should be between the corresponding planes of the loading outer ring raceway contacting with the ball as shown in Figure 5 – 18. The bearing manufacturer should supply this data.

**Figure 5 – 18 Correct transducer axial position**

The selectable transducer axial position: excepting for the deep groove ball bearings, the transducer axial position should be the outer ring width center. As shown in Figure 5 – 19. (this perhaps can cause the different vibration signal).

After selecting the transducer measuring point, the maximum permissible deviation is as:

For outer ring diameter $\leq 70\text{mm}$: $\pm 0.5\text{mm}$

For outer ring diameter $> 70\text{mm}$: $\pm 1.0\text{mm}$

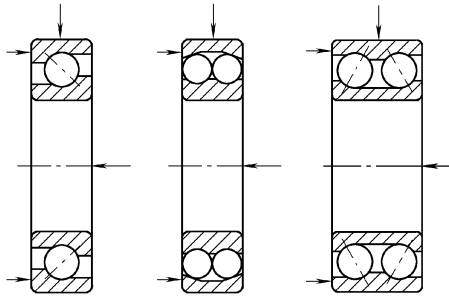


Figure 5-19 Selectable transducer axial position

Measuring direction: perpendicular to the rotating axis, as shown in Figure 5-20. The deviating radial direction deviation in any direction should be not over 5° .

5) Mandrel

For the mandrel cylindrical surface for mounting bearing inner ring, its outer diameter should be manufactured according to the f5 grade tolerance of ISO 286-2 and the geometric deviation should be the minimum value, in order to guarantee mounting the mandrel in slide fit into the bearing bore.

(4) Requirement for the operator

The qualified operator should guarantee to perform the bearing vibration measurement according to ISO 15242-2 stipulation.

4. Non-centralization (eccentricity) measuring for outside acting load

In the international standard ISO 15242-2, there is a special appendix as reference for explaining the non-centralization measuring method.

The loading tool displacement measuring is performed by using two dial gauges as shown in Figure 5-21. The dial gauges should be mounted on the spindle attachment and there should be certain distance between two dial gauges. The spindle should be slowly rotating, the loading pole radial

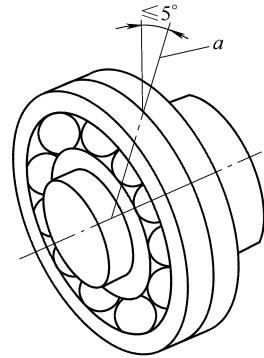
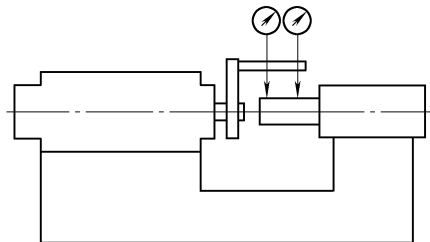


Figure 5-20 Transducer measuring direction deviation with radial direction axis

direction runout is measured.

It should be performed to correct the loading pole radial direction runout value with the relative measured bearing position.



**Figure 5 – 21 Non-centralization (eccentricity) measuring
for outside acting load**

Chapter 6 Varied Sliding Friction and Lubrication in Rolling Bearings

6.1 Varied Sliding Friction in Rolling Bearings

In a rolling bearing, while the rolling element is rolling forward under the applied load, the material ahead will be compressed along the rolling direction, therefore, the stress increases as shown on the left side curve of Figure 6 - 1, but the material behind will gradually recover to the original condition as shown on the right side curve of Figure 6 - 1.

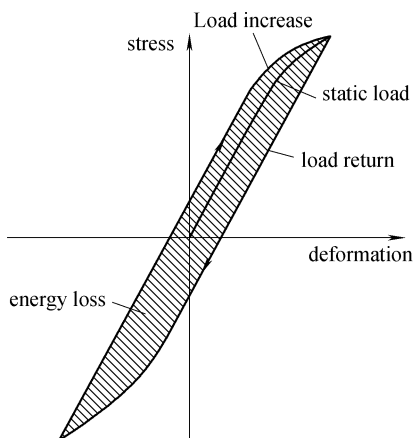


Figure 6 - 1 Elastic material residual elasticity hysteresis circle caused by repeatedly acting stress

It can be seen from Figure 6 -1 that the produced deformation value caused by the stress while the load increases is less than the produced deformation value caused by the stress while the load decreases. Therefore, there forms a residual elasticity hysteresis circle. The area included by the residual elasticity hysteresis circle represents the energy loss. That is this kind of energy loss is the friction loss during rolling contact. But comparing

with other types of friction loss in rolling bearing, this kind of friction loss caused by the residual elasticity hysteresis is very small.

The rolling bearing friction condition is very complicated. There are many kinds of sliding friction in rolling bearing. The following part will introduce varied sliding friction phenomena.

6.1.1 Differential Sliding Within Elastic Contacting Area

There is forming a contacting area in a rolling bearing when the rolling element is pressing on the raceway by the load. As shown in Figure 6-2, this contacting area is a spacial curve, which radius is not the rolling element radius, and is not the raceway curve radius as well. According to the Hertz theory, this curve radius is:

$$R = \frac{D_i D_w}{D_i + D_w}$$

In the equation:

D_i — raceway diameter in contacting area;

D_w — rolling element diameter in contacting area.

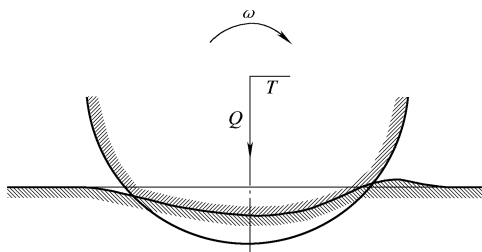


Figure 6-2 Contacting area between rolling element and raceway

The instantaneous rotating center is located where the relative velocity between the rolling element and the raceway is zero. That is, the rolling element linear velocity is as same as the raceway linear velocity at the instantaneous rotating center both in the direction and in the value.

The actual contacting area is a spacial curve surface. Therefore, there are only two points intersecting with the instantaneous rotating center in the contacting area, thus only these two points are the pure rolling points, but at all of other points of the contacting area, there are relative sliding

produced between the rolling element and raceway. In fact, the friction in the rolling bearing mainly is the sliding friction.

Figure 6-3 is a sketch of instantaneous rotating center and differential sliding direction.

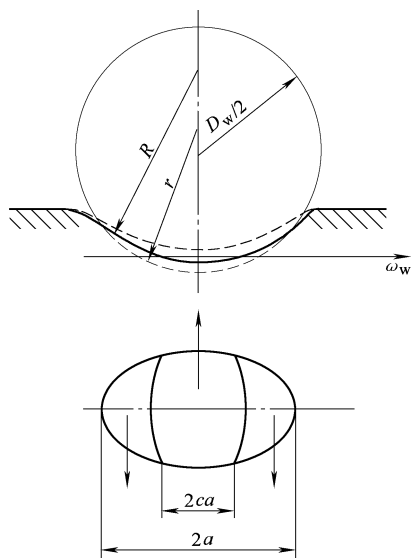


Figure 6-3 Instantaneous rotating center and differential sliding direction

D_w —rolling element diameter r —raceway groove curve radius

ω_w —Instantaneous rotating center axis $2a$ —contacting ellipse semi-major axis

$2ca$ —distance between two rolling motion lines

R —actual contacting area curve radius between rolling element and raceway

On the upper part of the actual contacting area intersecting with the instantaneous rotating center line, the rolling element linear velocity is smaller than the raceway contacting point linear velocity, but on the lower part of the actual contacting area intersecting with the instantaneous rotating center line, the rolling element linear velocity is larger than the raceway contacting point linear velocity. Therefore, as shown in Figure 6-4, all contacting area is divided as three parts having different sliding velocity direction.

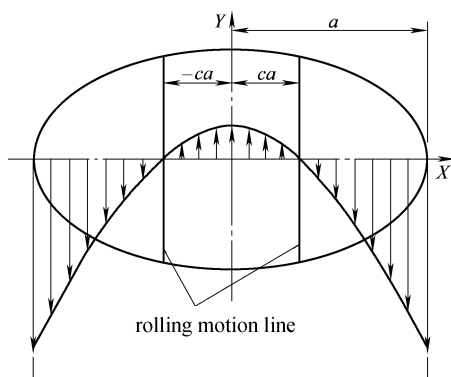


Figure 6-4 Differential sliding velocity direction in the contacting area

The differential sliding concept in rolling contact area has been put forward by Heathcote in 1920 in his published paper. Later on, many papers have verified that the differential sliding exists. D. TABOR verified by experiment in his famous paper that the differential sliding exists in the contacting area. The evidences are as follows:

1) Drilling a small hole on a ball and putting some ink in the hole, and making the ball rolling on the ready — prepared slot. If there is no relative slide, the ink trace should be very uniform. But the testing observing result is that the ink trace has been smeared very seriously. Therefore, it is deduced that there exists the relative sliding in the contacting area.

2) During the rolling, the rolled distance was measured after one week rolling. It is discovered that the measured rolled distances at different heights in the contacting area are shorter than the circle circumference length.

3) The measured friction value while there exists the differential sliding is ten times larger than the friction value while there is no differential sliding. According to the known theory and the experiment analysis, the rolling friction can not be reduced by the lubrication oil. But when lubrication by the lubrication oil, it is found that the measured friction value could reduce to a third of the original friction value. Therefore, it can be

confirmed that there exists the differential sliding.

Knowing from the rolling element force equilibrium, the instantaneous rotating center axis is not at the lowest point of the contacting area, but intersects with the contacting area at two points. The plane including the instantaneous rotating center line and parallel to the rolling direction intersects with the contacting curve surface such that the all intersection points form two curve lines. There are no relative sliding on these two curve lines. In all of the three regions divided by these two curve lines, there are relative sliding. The relative sliding direction of the middle part is in the contrary direction with the relative sliding direction of the two side parts. The distance between these two lines on which there are no relative sliding can be calculated out by analyzing the forces on the rolling element ball. Generally, the following value can be adopted:

$$C = 0.348$$

The differential sliding in the elastic contacting area is related to the ratio of the raceway groove curvature radius to the ball radius (that is : f_0 and f_i).

The more the raceway groove curvature radius is closed to the ball radius, the larger the differential sliding is. In high speed bearing, the differential sliding velocity can be so high that the local burn could occur.

6.1.2 Spinning Sliding

In an angular contact ball bearing, the ball contacts with the inner ring and the outer ring at the place of the contact angle α , in the contacting area, the linear velocity of the ball is different than the ring velocity, therefore, the spinning sliding can be formed. Figure 6-5 shows the principle of the spinning sliding.

In Figure 6-5, the ball does not contact with the inner ring and the outer ring at one point, but there forms a contacting area on the contact place. The ball linear velocity at every point in the contacting area is different than the ring raceway linear velocity, thus, the relative sliding can be formed.

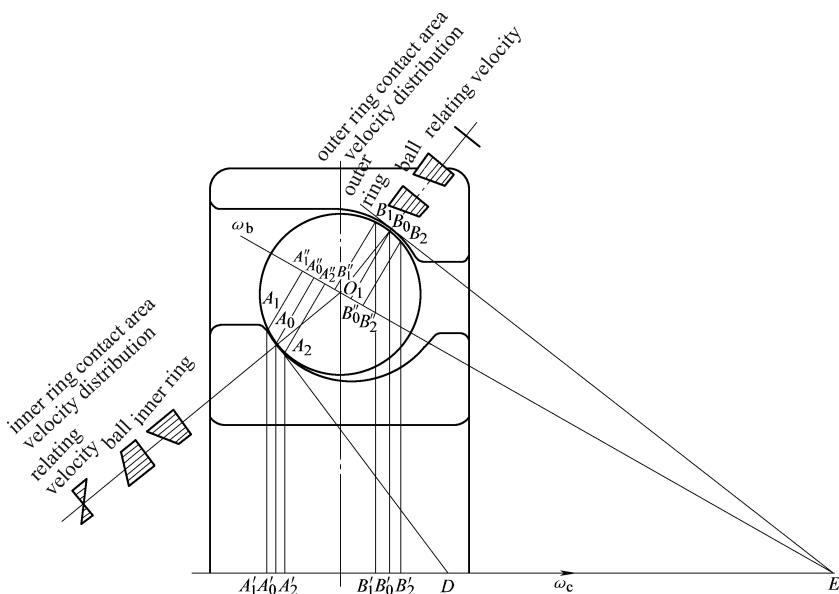


Figure 6-5 Spinning sliding principle

In Figure 6-5, $B_1 B_2$ is the contacting area of the ball with the outer ring, $A_1 A_2$ is the contacting area of the ball with the inner ring. It is assumed that the inner ring rotates in rotation speed n_i , the outer ring is stationary, and the ball rotates in angular velocity ω_b .

If adding an angular velocity ($-\omega_c$) to every element in the bearing, then, the relative motion relationship between every element will keep not changed. At this time, the ball center is in stationary, but the outer ring rotates in ($-\omega_c$), the ball rotation direction is determined according to different condition.

In Figure 6-5, the E point is the intersection point of a tangential line making from the outer ring contacting area with the bearing center line, the D point is the intersection point of a tangential line making from the inner ring contacting area with the bearing center line. If assuming that the ball rotation axis also intersects with the bearing center line at the E point, then the ball rotation axis is the ω_{b0} direction as shown in Figure 6-5. If the B_0 point velocity on the ball is equal to the B_0 point velocity on the outer ring

raceway, then the distance of every point on the outer ring raceway between B_1 and B_2 to the bearing center line, $B_1 B'_1$ and $B_2 B'_2$, gradually decreases. Therefore, the every point velocity also gradually decreases. But the distance of every point on the ball between $B_1 B_2$ to the ball rotating axis EO_1 , $B_1 B''_1$ and $B_2 B''_2$, also gradually decreases. Therefore, the every point velocity in the ball contacting area also gradually decreases. It has been known from the geometry relation that there is no relative sliding produced in this condition, which velocity changes are shown in the upper part of Figure 6-5.

However, the contacting condition of the ball with the inner ring is just the contrary condition, The distance of every point in the contacting area $A_1 A_2$ to the bearing center line, $A_1 A'_1$ — $A_2 A'_2$, gradually decreases. Therefore, the every point velocity of the inner ring raceway also gradually decreases. But the distance of every point in the ball contacting area within the A_1 point to the A_2 point to the the ball rotating axis EO_1 , $A_1 A''_1$ and $A_2 A''_2$, gradually increases. Therefore, the every point velocity in the ball contacting area gradually increases. Therefore, on the every point of the ball contacting area with the inner ring raceway, there is relative sliding produced, as shown in the lower part of Figure 6-5. At last, the spinning sliding is formed.

If the ball rotating axis intersects with the bearing center line at the D point, it can be deduced that the spinning sliding is produced on the outer ring contacting area, but the spinning sliding is not produced on the inner ring contacting area. If the ball rotating axis intersects with the bearing center line at any point within the D point to the E point, then the spinning sliding is produced both on the outer ring contacting area and the inner ring contacting area.

The ball rotating axis can only intersect with the bearing center line at one point, therefore, there would be two conditions:

- 1) Inner ring controlling: the spinning sliding is produced on the outer ring contacting area. This condition is called as inner ring controlling.
- 2) Outer ring controlling: the spinning sliding is produced on the inner

ring contacting area. This condition is called as outer ring controlling.

As to in the actual rolling bearing, within which ring contacting area could the spinning sliding be produced ?

It is required to calculate actually the friction moment for every ring raceway contacting area. Usually, the spinning sliding could be produced on the ring raceway contacting area with smaller friction moment. The following two deductions can be got from many references:

1) At low rotation speed, the ball centrifugal force is very small, so it can be omitted. The acting force between the ball and the inner ring or the outer ring contact area is the same. But usually, the contacting area between the ball and the inner ring raceway is larger than the contacting area between the ball and the outer ring raceway, so the resisting friction moment for the spinning sliding is larger. Therefore, the spinning sliding is produced on the contacting area between the ball and the outer ring raceway.

2) At high rotation speed, the ball centrifugal force is very large, the acting force on the contacting area between the ball and the outer ring raceway is larger than that on the inner ring raceway contacting area, so the resisting friction moment for the spinning sliding is increased. Therefore, the spinning sliding is produced on the inner ring contacting area.

6.1.3 Gyroscopic Sliding

In an angular contact bearing, because the ball implication motion is a circumferential motion, while the ball self-rotation axis direction is always changeable, the ball is subjected to an inertia moment, or called gyroscopic moment. This inertia moment expression deduced in reference [25] is as follows:

$$\begin{aligned} M_k &= \frac{8}{15} \frac{\pi \gamma}{g} \left(\frac{D_w}{2} \right)^5 \omega_c \omega_b \sin \alpha \\ &= J_0 \omega_b * \omega_c \end{aligned}$$

where γ — ball material specific gravity;

g — gravity acceleration;

D_w — ball diameter;

ω_b — ball self rotation angular velocity;

ω_c — ball self revolution angular velocity;

M_k — inertia moment;

J_0 — moment of inertia of the ball.

Because both ω_b and ω_c are proportional to the bearing inner ring rotation speed, the inertia moment M_k is proportional to the square of the bearing inner ring rotation speed. Thus, the inertia moment M_k value is very large in the high speed bearing. Under low speed running, in order to prevent the gyroscopic sliding, it is always used to increase the axial force for increasing the contacting area friction moment. As mentioned in section 4.5.

In high speed bearing, the gyroscopic sliding could occur when the inertia moment M_k value is higher than the contacting area friction moment value.

6.1.4 Sliding Friction Related to Cage

In rolling bearing, there are three cage guiding modes: rolling element guiding, inner ring guiding and outer ring guiding, as shown in Figure 6-6.

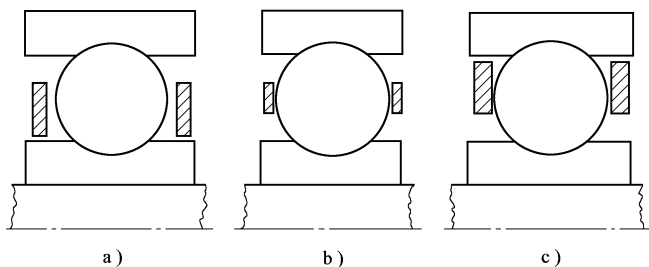


Figure 6-6 Cage guiding modes

a) rolling element guiding b) inner ring guiding c) outer ring guiding

Sliding friction occurs in all of the three cage guiding modes. There is sliding friction happening between the cage bore and the rolling element.

In the loading area, the rolling element pushes the cage rotating; but in the non-loading area, the cage pushes the rolling element rotating; in these two cases, both have sliding friction happening. Within the inner ring guiding and the outer ring guiding, there also is sliding friction happening between the cage and the inner ring outside diameter surface or the outer

ring inner diameter surface.

6.1.5 Sliding Friction Between Roller ends and Ring Rib

Between the tapered roller bearing large end face and the inner ring rib, between the asymmetrical spherical roller large end face and the ring rib, between the cylindrical roller bearing end face subjecting to axial load and the ring rib, there exists the sliding friction.

6.1.6 Sliding Friction Related to Sealing Ring

In the sealing bearing with elastic seals, between the seals and the contacting surface, there also exists the sliding friction, especially between the contacting type seals and the contacting ring surface.

6.2 Rolling Bearing Friction Moment and Friction Coefficient

6.2.1 Rolling Bearing Friction Moment

The Rolling bearing friction moment is the resisting bearing running moment, which is produced by the total value of the rolling friction value, the sliding friction value and the lubrication friction value, expressed by M . According to a large number of testing data of SKF, A. Palmgren divided the total rolling bearing friction moment as two parts: M_1 which is related to the bearing load and M_0 which is not related to the bearing load, that is:

$$M = M_0 + M_1$$

(1) Calculation of M_0

In the high speed and light load applications, M_0 plays the main function role, which is mainly related to the bearing type, the lubrication viscosity and quantity, and the bearing rotation speed.

While $\nu n \geq 2000$

$$M_0 = 10^{-7} f_0 (\nu n)^{2/3} D_m^3$$

While $\nu n < 2000$

$$M_0 = 160 * 10^{-7} f_0 D_m^3$$

Here M_0 — friction moment that is not related to the bearing load

(N. mm) ;

D_m — bearing mean diameter (mm) ;

f_0 — factor related to the bearing type and the lubrication , listed in Table 6 - 1 ;

n — bearing rotation speed (r/min) ;

ν — lubrication kinematic viscosity under the bearing working condition (for grease , taking the base oil viscosity) (mm²/s).

For light series bearings, lower value will be taken; for heavy series bearings, higher value will be taken.

Table 6 - 1 Factor f_0

Bearing type	oil mist lubrication	oil bath lubrication or grease lubrication	oil bath lubrication (vertical shaft) or spra oil lubrication
deep groove ball bearing	0. 7 ~ 1	1. 5 ~ 2	3 ~ 4
aligning ball bearing			
angular contact ball bearing			
single row	1	2	4
double row	2	4	8
radial cylindrical roller bearing			
with cage	1 ~ 1. 5	2 ~ 3	4 ~ 6
full complement	—	2. 5 ~ 4	—
aligning roller bearing	2 ~ 3	4 ~ 6	8 ~ 12
tapered roller bearing	1. 5 ~ 2	3 ~ 4	6 ~ 8
thrust ball bearing	0. 7 ~ 1	1. 5 ~ 2	3 ~ 4
thrust cylindrical roller bearing	—	2. 5	5
thrust aligning roller bearing	—	3 ~ 4	6 ~ 8

(2) Calculation of M_1

M_1 is mainly the friction loss of the elastic material residual hysteresis and the contacting surface differential sliding, calculated by the following

equation:

$$M_1 = f_1 P_1 D_m$$

Here f_1 — factor related to the bearing type and the load, listed in Table 6-2;

P_1 — bearing load while calculating bearing friction moment.

Table 6-2 f_1 and P_1 calculation equation

Bearing type	f_1	P_1 ^①
deep groove ball bearing	$0.0009 (P_0/C_0)^{0.55}$	$3F_a - 0.1F_r$
aligning ball bearing	$0.0003 (P_0/C_0)^{0.4}$	$1.4Y_2 F_a - 0.1F_r$
angular contact ball bearing		
single row	$0.0013 (P_0/C_0)^{0.33}$	$F_a - 0.1F_r$
double row	$0.001 (P_0/C_0)^{0.33}$	$1.4F_a - 0.1F_r$
aligning roller bearing	0.0005	$1.2Y_2 F_a$
radial cylindrical roller bearing		
with cage	$0.00025 \sim 0.0003$ ^②	F_r ^③
full complement	0.00045	F_r ^③
tapered roller bearing	$0.0004 \sim 0.0005$ ^②	$2YF_a$
thrust ball bearing	$0.0012 (P_0/C_0)^{0.33}$	F_a
thrust cylindrical roller bearing	0.0018	F_a
thrust aligning roller bearing	$0.0005 \sim 0.0006$ ^②	$F_a (F_{rmax} \leq 0.55F_a)$

- ① If $P_1 < F_r$, taking $P_1 = F_r$.
 ② For light series bearings, taking smaller value; for heavy series bearings, taking larger value.
 ③ For short cylindrical roller bearings, if subjecting to axial load, it should be considered to add the additional moment M_2 .

In Table 6-2:

- P_0 — static equivalent load (N);
 C_0 — basic rating load (N);
 F_a — axial load (N);
 F_r — radial load (N);

Y, Y_2 — axial factor, found from the bearing catalog.

(3) Calculation of M_2

It should be considered that if a short cylindrical roller bearing are subjected to the radial load and the axial load at the same time, the additional friction moment M_2 should be added, that is the total bearing friction moment is:

$$M = M_0 + M_1 + M_2$$

$$M_2 = f_2 F_a D_m$$

Here f_2 — factor related to the bearing type and the lubrication mode.

Listed in Table 6 - 3. In Table 6 - 3, the f_2 value is suitable for $K_v = \nu / \nu_1 = 1.5$ and $(F_a / F_r) \leq 0.4$. In the K_v equation:

ν — lubrication oil kinematic viscosity under the bearing working condition (mm^2/s);

ν_1 — the minimum lubrication oil kinematic viscosity under the bearing working condition that is essential for forming the lubrication oil film (mm^2/s).

Table 6 - 3 Factor f_2

Bearing structure	Oil lubrication	Grease lubrication
with cage	0.006	0.009
full complement	0.003	0.0006

6.2.2 Rolling Bearing Friction Coefficient

It is common used to express the friction value in a rolling bearing by the friction coefficient μ , which definition is as:

$$\mu = \frac{2M}{Pd}$$

Here μ — rolling bearing friction coefficient;

d — bearing bore diameter (mm);

M — bearing total friction moment ($\text{N} \cdot \text{mm}$);

P — bearing equivalent dynamic load; for radial bearing, that is the radial load; for thrust bearing, that is the axial load.

Table 6-4 lists the friction coefficient for every type rolling bearing, which has a certain changing range. When $P / C \leq 0.05$, larger value of friction coefficient should be taken; when $P / C \geq 0.1$, smaller value of friction coefficient should be taken (C — bearing rating dynamic load). It is very clear from the table values that the deep groove ball bearing, the self-aligning ball bearing and the single row cylindrical roller bearing are suitable for using in applications for low friction coefficient requirement.

Table 6-4 Rolling bearing friction coefficient

Bearing type	μ
deep groove ball bearing	0.0015 ~ 0.0022
aligning ball bearing	0.0010 ~ 0.0018
single row cylindrical roller bearing	0.0011 ~ 0.0022
aligning roller bearing	0.0018 ~ 0.0025
needle roller bearing	0.0025 ~ 0.0040
angular contact ball bearing	0.0018 ~ 0.0025
tapered roller bearing	0.0018 ~ 0.0028
thrust ball bearing	0.0013 ~ 0.0020
thrust aligning roller bearing	0.0018 ~ 0.0030

The rolling bearing friction moment can be divided as the dynamic friction moment produced by the friction load in the rotating motion and the starting friction moment caused by the friction moment at the beginning of running from the stationary condition. Usually, the starting friction moment is larger than the dynamic friction moment in motion. The friction moment is the rolling bearing comprehensive using performance that is the result produced by the composition of many factors. There are not only rolling friction and sliding friction, but also the friction between the solid bodies and the lubricant viscous resisting friction as well. From the engineering application considerations, it is required to know how much the friction moment value is for a given type and dimension bearing, under certain load and rotating speed. Therefore, the rolling bearing friction moment

calculation method has been gradually developed based on the experienced constants (detailed in section 6.2.1 of this chapter). It is also required to illustrate that the starting friction moment could be much larger than the calculation value. In some cases, for considering the contacting seals friction moment, it is required to add certain value to the M_0 value.

The calculated bearing friction moment value usually is just the approximate value under the normal conditions. When it is required to know the precise bearing friction moment value for a given condition, the actual measurement should be performed.

6.3 Rolling Bearing Lubrication

6.3.1 Function of Lubrication and Lubricant Selection

For assuring rolling bearing normal running, there must be good lubrication. The rolling bearing rating load and the rotation speed limit both are determined under the assumption that there must be proper lubrication. So called proper lubrication designates that the lubricant selection is right, the lubricant quantity is appropriate. The incorrect lubrication is one of the main reasons causing bearing earlier damage.

There are many kinds of sliding in rolling bearing that will cause the frictional heat and the wear. The main purpose for lubricating rolling bearing is to prevent from the direct metal to metal contacting between the surfaces of the raceway, the rolling element and the cage for reducing the frictional heat, preventing from high temperature rise and reducing the bearing component wearing, preventing from corrosion. When circulating oil lubrication, the oil flowing can take the role of dissipating heat; when using grease lubrication, the grease can also take the role of sealing.

Table 6 – 5 lists the considering factors for selecting the lubricant. Table 6 – 6 lists the recommended $D_m n$ value for the oil lubrication and the grease lubrication. Table 6 – 7 lists the working temperature range suitable for the grease.

Table 6 – 5 Lubricant selection principle

According to		selection principle
bearing speed		The higher the bearing rotation speed is, the larger the frictional heat is. In the high speed applications, the low viscosity lubrication oil or the high degree penetrating grease should be selected. In the low speed applications, that is contrary. The recommended $D_m n$ values for the grease lubrication and oil lubrication are listed in Table 6 – 6.
working temperature		During rolling bearing running, because of the frictional heat, the bearing temperature will be raised. Every type lubricant has its own suitable temperature range; besides, the temperature also is the factor of influencing the bearing accuracy. Therefore, in high temperature applications, the large viscosity and high flash point oil or the small degree working awl, high dripping point and heat-resistant grease should be selected. The recommended temperature range suitable for the grease is listed in Table 6 – 7.
bearing load		The bearing load value influences the formation of the oil film in a large scale, the heavier the load is, the more difficult the oil film formation is. Therefore, in the heavy load applications, the large viscosity oil or the small degree penetrating grease should be selected. In the light load applications, that is contrary. In the shocking load application, the larger viscosity oil or the lower penetrating degree grease should be selected.
surroundings		If the surroundings air is moist, there is more dust, and the sealing device is too simple, then the calcium base grease that is not easily dissolving to water should be selected. If there is just little water, the lithium base grease should be selected.
installation condition		For the bearings installed on a vertical or inclined shaft, the lubricant is easily lost, excepting for special caring to the sealing device, it should select the larger viscosity oil or the lower penetrating degree grease.
lubrication mode	dropping oil	selecting lower viscosity oil
	circulating oil	selecting lower viscosity oil
	spray oil	selecting the oil with resisting oxide

Table 6 – 6 Recommended $D_m n$ value for oil lubrication and grease lubrication

Bearing type	Grease lubrication	Oil lubrication			
		bath oil	dropping oil	mist , oil air	spray oil
deep groove ball bearing	300000	500000	600000	1000000	2500000
angular contact ball bearing	300000	500000	500000	900000	2500000
cylindrical roller bearing	300000	400000	400000	1000000	2000000
tapered roller bearing	250000	350000	350000	450000	—
thrust ball bearing	70000	100000	200000	—	—

- Note: 1. D_m — bearing mean diameter, that is (outer diameter + bore diameter) /2 (mm) ; n — shaft rotation speed (r/min) .
2. $D_m n$ value for bath oil , dropping oil , mist , oil air , spray oil is suitable for high speed and high accuracy bearing. For subjecting heavier load bearing , 85% value should be taken.

Table 6 – 7 Recommended temperature range suitable for grease

Grease type	Suitable working temperature/℃
lithium base grease	– 30 ~ + 110
lithium base compound grease	– 20 ~ + 140
sodium base grease	– 30 ~ + 80
sodium base compound grease	– 20 ~ + 140
calcium base grease	– 10 ~ + 60
calcium base compound grease	– 20 ~ + 130
alumina compose grease	– 30 ~ + 110
boron base compose	– 20 ~ + 130
polyester grease	– 30 ~ + 140

6. 3. 2 Lubricant Type and Main Performance

According to lubricant shape , lubricant can be divided as three kinds: lubrication oil , lubrication grease and solid lubricant.

1. Lubrication oil

The lubrication oil mainly designates the mineral oils specially produced by the oil plant , secondly the plant oil , compound oil. When the bearing

runs in high speed or works in high temperature condition, and the grease lubrication can not be used, or the neighboring machinery component (such as gear) is lubricated by the oil, the bearing is always lubricated by oil. The main characteristics of lubrication oil quality are listed in Table 6 – 8.

**Table 6 – 8 Main lubrication oil quality norm meaning
and using significance**

Characteristics	Meaning	using significance
viscosity	expresses the relative thick or thin degree; expresses the internal frictional force between fluid elements when the fluid flows by applied force.	The viscosity is the main basis for selecting the lubricated oil. The viscosity value is related to the oil film being or not being forming under certain pressure and temperature, to the oil putting into or not into two frictional surfaces and to the oil film thickness between two friction surfaces directly. While the viscosity value is large, and the pressure is large, it is not easy to squeeze the oil out and can keep certain thickness of oil film, but it is not easy to put the oil into the narrow fit clearance between the friction surfaces; it is the contrary for the lower viscosity.
viscosity index	viscosity index is an oil index that can reflect the viscosity change with the temperature change.	Usually, it is hope to have large viscosity index, that is when the temperature is rising or lowering, the viscosity changing is less. The viscosity-temperature characteristic is better, the stable lubrication condition can be kept.
solidifying point	The maximum temperature when the fluid is cooled to the point losing fluidity under test condition.	expressing the lubrication oil cold resistant performance
acid value	Required hydroxide (KOH) mg value for neutralization of the 1g lubrication oil	The acid value is one of the important characteristics for controlling and reflecting the lubrication oil making degree, that is also the characteristic for judging the oil wasting degree

(continued)

Characteristics	Meaning	using significance
flash point	The lowest temperature when heating the oil to the flashing light beginning under the serious stipulation condition. According to the measuring instrument, there are the open flash point and the close flash point.	The flash point is an important norm for petroleum production evaporating and safety, also can judge if there is residual light distillate or mixing in fuel oil
ash content	The burned black residual thing formed by lubrication oil evaporated and heated	The more residual charcoal oil can always block the oil path, increase wearing. The residual charcoal oil can not be selected as the lubrication oil for the precise machine tool
ash content	The residual solid material weight percentage after burning at the stipulation temperature	The main material in the ash content is the metal oxide, which exists in the lubrication oil in salt or oxide condition. For the lubrication oil without additive, the ash content designates the precise making degree. The lower the ash content is, the higher the precise making degree is and the more stable the lubrication oil is. Because most of the additives are the metal salts, for the lubrication oil with additive, more ash content is permissible.
machinery impurity	The suspended or precipitated dust, sand and metal particles in the lubrication oil that are not dissolving in gasoline and benzene	The machinery impurity can destroy the oil film, block the oil path and the filter, accelerate machine wear. The normal lubrication oil requires that there is not including the machinery impurity or just including very little machinery impurity

The viscosity is the main performance norm of lubrication oil. The viscosity value is directly influencing the lubrication oil fluidity and the forming oil film thickness between the frictional surfaces. Table 6-9 shows

several expressing methods and the measuring unit for the lubrication oil viscosity.

Table 6-9 Several expressing method and the measuring unit for the lubrication oil viscosity

Name	Legal measuring unit		not-legal measuring unit		Unit conversion
	name	symbol	name	symbol	
(dynamic) viscosity	Pascal-seconds	Pa · s	Poise	P, P _o	1P = 10 ⁻¹ Pa · s
			Centipoise	cP	1cP = 10 ⁻³ Pa · s
kinematic viscosity	quadratic meter per second	m ² /s	stoke	St	1St = 10 ⁻⁴ m ² /s
			Centistoke	cSt	1cSt = 10 ⁻⁶ m ² /s

2. Grease lubrication

Lubrication greases are made by the lubrication oil and the grease thickener and the additive (some times without additive) mixing in high temperature. Lubrication greases are mainly divided as calcium base grease, sodium base grease, calcium an sodium base grease, lithium base grease, alumina base grease and molykote grease etc.

The main lubrication grease quality norms are the penetrating degree, the dripping point, the machine stability, the oxidization stability, the anti-corrosive performance and so on. Table 6-10 shows the main lubrication oil quality norm meaning and using significance.

Table 6-10 Main lubrication oil quality norm meaning and using significance

Norm	Meaning	Using significance
dripping point	The dripping point is the temperature at which temperature the grease begins dripping the first drop oil changing the non flowing condition to the flowing condition when put the grease into the dripping meter.	The dripping point is a reference norm for measuring the grease heat-resistant feature. The highest application temperature should be 20 ~ 30℃ lower than the grease dripping point

(continued)

Norm	Meaning	Using significance
working penetrating degree	The penetrating degree is the sinking depth in heating to 25℃ grease sample in 5 seconds. The unit is 1 /10mm. The penetrating degree is an identification norm for the grease being thick or thin degree (the old name is needle penetrating degree)	While the penetrating degree is smaller, the plasticity is larger, the fluidity is less, the pumping ability is also less; while the penetrating degree is larger, that is the contrary. Besides, after grease shearing, the grease consistence is changeable. The grease machinery stability can be known by measuring the penetrating degree difference before and after sharing.
machinery impurity	The included matter value in the grease, such as the dust, the sand particles and the metal cutting piece and so on	The machine component can produce wearing if the machinery impurity is mixing in.

3. Solid lubricant

On the special applications, owing to varied reasons, when the lubrication oil and the lubrication grease usage are limited, the solid lubricant can be selected. There are three methods:

- 1) Adding the solid lubricant into the lubrication grease. Usually, it is the way to add 3% ~ 5% No.1 molykote into the lubrication grease for reducing the wear and raising the pressure resisting and the heat resisting ability.
- 2) The solid lubricant is adhered onto the surfaces of the raceways and the cage or the rolling elements by the adhesives. In this way the thin lubrication film can be formed, the friction and wear can be reduced in some extent.
- 3) The bearing components are made by the engineering plastic or the powder metallurgy adding the solid lubricant for having the good self lubrication performance. These bearings can be used in the special working conditions, such as the high temperature condition, the high pressure

condition, the high vacuum condition, the corrosion-resisting condition, the anti-radiating condition and the extreme low temperature condition. There is much better result if matching these self lubrication material with the thin solid lubrication film.

6.3.3 Grease Lubrication

1. Grease selection

Usually, for the lubrication grease selection, the grease type and the grease performance are determined according to the bearing working condition, the working temperature and the load condition.

1) Selecting the grease based on the working temperature. The main norms are the dripping point, the oxide stability, and the low temperature performance. The bearing actual working temperature should be $10 \sim 20^{\circ}\text{C}$ lower than the grease dripping point. For the compound grease, the bearing actual temperature should be $20 \sim 30^{\circ}\text{C}$ lower than the grease dripping point.

2) Selecting the grease based on the bearing load condition. For the heavy load, the low working penetration degree grease should be selected. For working in high pressure condition, excepting the low working penetration degree, the grease with high oil film strength and the extreme pressure (EP) ability should be selected.

3) Selecting the grease based on the environmental conditions, The Calcium base grease is not dissolving in water, which is used in the more moisture content applications and the wet applications. The sodium base grease is easy dissolving in water, which is suitable for the less moisture and dry applications.

Table 6 – 11 lists the commonly used grease performance.

Table 6 – 12 lists the commonly used machine tool rolling bearing grease main quality norm and usage.

2. Filling grease amount and grease re-lubrication interval

Filling grease amount in a bearing should be proper. If the filling grease amount is undue excessive, the bearing running temperature would be increased; at the same time, there would be no necessity loss.

Table 6 – 11 Commonly used rolling bearing grease performance

	Calcium base (Ca)	complex Calcium base	Sodium base Na		Alumina base Al	Calcium-lead base Ca-Pb
			long fiber	short fiber		
appearance	emulsion paste	emulsion paste	light fiber	emulsion past	transparent draw string	short fiber emulsion paste
color	yellow	brown	yellow- blue	yellow- blue	blue-red	yellow- light yellow
dripping point /℃	70 ~ 90	280 ~ 280	140 ~ 200	140 ~ 200	70 ~ 90	70 ~ 180
low temperature starting torque	middle-worse	middle-worse	high	worse	middle	worse
water resistant	good	good	worse	worse	better	good
heat resistant	low	good	OK	OK	low	OK
machinery stability	OK	good	good	better	OK	better
pumping ability	OK	OK	bad	good	OK	good
suitable temperature /℃	- 10 ~ + 70	- 10 ~ + 150	- 10 ~ + 100	- 10 ~ + 100	- 10 ~ + 80	- 10 ~ + 90
suitable speed	middle-low	middle-low	middle-low	high-low	middle-low	middle-low
resisting shock load ability	no	OK	OK	OK	OK	OK
hand lubrication	OK	OK	OK	OK	OK	OK
grease cup lubrication	OK	OK	OK	OK	OK	OK
grease shorting spear lubrication	OK	OK	no	no	OK	OK
concentrating lubrication	no	no	no	no	OK	OK
usage	middle and low speed and possibly meeting water and the wet position bearing	suitable for sliding and roll- ing bearings	middle and low speed bearings	high tem- perature rolling bearings	subjected to vibration bear- ings and car ch- assis bearings	rolling mill shocking posi- tion bearings
note	good water- proof seal, water content 1. 5% ~ 3%	can add EP additive using in high temper- ature, good water- proof	including no water content preventing from emulsion	including no water content preventing from emulsion	EP good , good stability under dripping point. not rest- ore after cool above the drip- ping point	EP good

(continued)

	Barium base (Ba)	Calcium- sodium base (Ca-Na)	Lithium base Li		complex lithium base	non-soap base (non-dissolve)
			petroleum	compound		
appearance	short fiber paste	short fiber paste	emulsion paste	short fiber paste	emulsion paste	emulsion past
color	red- yellow	yellow-green	brown-red	brown-red	brown-yellow	yellow-red
dripping point /℃	150 ~ 180	150 ~ 180	170 ~ 190	170 ~ 220	200 ~ 280	no
low temperature starting torque	middle	middle-low	middle-low	low	low	middle-low
water resistant	good	OK	good	good	good	good
heat resistant	OK	OK	good	good	good	good
machinery stability	OK	OK	good	good	good	good
pumping ability	OK	OK	good	good	good	OK
suitable temperature /℃	- 10 ~ + 130	- 10 ~ + 120	- 30 ~ + 130	- 50 ~ + 130	- 30 ~ + 130	- 10 ~ + 200
suitable speed	middle-low	high-low	high-low	high-low	high-low	middle-low
suitable load	high-low	middle-low	high-low	high-low	high-low	middl-low
resisting shock load ability	OK	OK	OK	OK	OK	no
hand lubrication	OK	OK	OK	OK	OK	OK
grease cup lubrication	OK	OK	OK	OK	OK	OK
grease shorting spear lubrication	OK	OK	OK	OK	OK	OK
concentrating lubrication	OK	no	OK	OK	OK	OK
usage	rolling bearing	high speed rolling bearing	middle- small type rolling bearing	high-low temperature long life rolling bearing	high-low temperature long life rolling bearing	high temper- ature rolling bearing
note	large soap content not suit for high and low temperature	with certain water resistance and the mach- inery stability				

Table 6 - 12 Commonly used machine tool rolling bearing grease main quality norm and usage

name	symbol	appearance	dripping point ≥/℃	working penetrating degree	water content (%)	ionize NaON(%)	ionize organic acid	dividing oil amount (pressure) ≤ (%)	machinery impurity (acid dividing)	feature and usage
Calcium base grease (GB 491— 1987)	ZG-1 ZG-2 ZG-3 ZG4	light yellow to dark brown un- iform oil paste	80 85 90 95	310 ~ 340 265 ~ 295 220 ~ 250 175 ~ 205	1. 5 2. 0 2. 5 3. 0					Good water resist- ance. not easy emu- lision; low drip point, will be lost over 60℃. suitable for normal bea- ring. ZG-1 is for central fill grease lower then 55℃. ZG2- for moder- ate speed. ZG-3-for lower than 60℃ moderate load, ZG4-for heavier load
Complex Calcium base grease (ZB E6003— 1988)	ZFG-i ZFG-2 ZFG-3 ZFG-4	light yellow- dark brown un- iform paste with- out lump	180 200 220 240	310 ~ 340 265 ~ 295 220 ~ 250 175 ~ 205	mark	0. 2 0. 2 0. 2 0. 2	no	13 10 7 5	no	higher temperature re- sistance. Good hot plas- ticity, good water resist- ance, suitable for higher temperature and load bearing

(continued)

name	symbol	appearance	dripping point $\geq/^{\circ}\text{C}$	working penetrating degree	water content (%)	ionize NaON(%)	ionize organic acid	dividing oil amount (pressure) $\leq (%)$	machinery impurity (acid dividing)	feature and usage
Lithium base grease(GB 7324— 1994)	ZL-1 ZL-2 ZL-3 ZL-4	light yellow- dark brown uniform paste	170 175 180 185	310 ~ 340 265 ~ 295 220 ~ 250 175 ~ 205	mark	0. 1 0. 1 0. 15 0. 15	no	14 12 10 8	no	high effective grease with oxidation resisting, good water resisting. suit- able for machine bearing
Complex Lithium base grease (SH 0380— 1992)	ZL-1H ZL-2H ZL-3H ZL-4H	light brown- dark brown uniform paste	170 175 180 185	310 ~ 340 265 ~ 295 220 ~ 250 175 ~ 205	mark	0. 1 0. 1 0. 15 0. 15	no	14 12 10 8	no	more usage grease grease with some water resisting, good better ma- chinery stability, suitable every type bearing
precise machine tool spindle bearing (SY1417— 1980)	No. 2 No. 3		180 180	265 ~ 295 220 ~ 250	mark	0. 1 0. 1			no	good oxidize stability with oxidizer resist, colloid stability, machinery sta- bility and rust-proof. mainl used for precise machine tool bearing.

(continued)

name	symbol	appearance	dripping point ≥/℃	working penetrating degree	water content (%)	ionize NaON (%)	ionize organic acid	dividing oil amount (pressure) ≤ (%)	machinery impurity (acid dividing)	feature and usage
No. 7018 high speed bearing grease (ZBE 40012— 1988)		yellow-light brown uniform paste	200	64 ~ 78 (1/4 penetrating degree)				10	(number/cm ³) diameter ≥ 0. 025mm , ≤ 1000 ; diameter ≥ 0. 075mm , ≤ 120 ; diameter ≥ 0. 125mm , no	
Molybdenum dioxide complex Calcium base grease	ZFG-1E ZFG-2E ZFG-3E ZFG-4E	gray black paste	180 200 220 240	310 ~ 350 260 ~ 300 210 ~ 250 160 ~ 200	0. 1 0. 1 0. 1 0. 1					Adds certain amount molybdenum in the Cal- cium base grease , good wear-proof. ZFG-1E suit automatic filling grease system under 55℃ ; ZFG- 2E suit for moderate speed and light load un- der 60℃ ; ZFG-3E suit for moderate speed and

(continued)										
name	symbol	appearance	dripping point ≥/℃	working penetrating degree	water content (%)	ionize NaON (%)	ionize organic acid	dividing oil amount (pressure) ≤ (%)	machinery impurity (acid dividing)	feature and usage
Molybdenum dioxide complex Calcium base grease	ZFG-1E	gray black paste	180	310 ~ 350	0. 1					moderate load under 65℃ ; ZFG-4E suit for low speed and heavier load under 70℃ .
	ZFG-2E		200	260 ~ 300	0. 1					
	ZFG-3E		220	210 ~ 250	0. 1					
	ZFG-4E		240	160 ~ 200	0. 1					
Molybdenum Calcium base grease	ZL-1E	gray black paste	175	310 ~ 340	no					Adds certain amount molybdenum in the Cal- cium base grease, good wear-proof. with EP performance. Suit for heavier lo load and high temperature application.
	ZL-2E		175	265 ~ 295						
	ZL-3E		175	220 ~ 250						
	ZL-4E		175	175 ~ 205						

Usually, for the having been running bearings, it is better to replenish the lubrication grease periodically (short interval). The replenishing grease amount can be estimated according to the following equation:

$$G = 0.005BD$$

In the equation G — replenishing grease amount (g);

D — bearing outer diameter (mm);

B — bearing width (mm).

For the bearings of new filling grease, the general principle is that to determine the filling grease amount according to the ratio of bearing rotation speed to the bearing rotation speed limit:

$n_i/n < 1.25$ the filling grease amount should be one third of the free space in a bearing;

$1.25 \geq n_i/n \leq 5$ the filling grease amount should be one third to two thirds of the free space in a bearing;

$n_i/n > 5$ the filling grease amount should be two thirds of the free space in a bearing;

Table 6 – 13 lists the recommended filling grease amount in JB/T 7752—1995 for the seal bearings.

Table 6 – 13 Non-contacting type sealing ball bearing filling grease amount^①

(unit: g)

bearing model	filling grease amount	bearing model	filling grease amount	bearing model	filling grease amount
6002-2RZ	0.24 ~ 0.6	6011-2RZ	3.6 ~ 9.0	6203-2RZ	0.6 ~ 1.5
6300-2RZ	0.36 ~ 0.9	6012-2RZ	3.96 ~ 9.9	6204-2RZ	0.88 ~ 2.2
6004-2RZ	0.56 ~ 1.4	6013-2RZ	4.2 ~ 10.5	6205-2RZ	1.04 ~ 2.6
6005-2RZ	0.68 ~ 1.7	6014-2RZ	5.7 ~ 14.2	6206-2RZ	1.64 ~ 4.1
6006-2RZ	0.96 ~ 2.4	6015-2RZ	5.9 ~ 14.8	6207-2RZ	2.44 ~ 6.1
6007-2RZ	1.32 ~ 3.3	6016-2RZ	8.1 ~ 20.2	6208-2RZ	3.2 ~ 8.0
6008-2RZ	1.64 ~ 4.1	6200-2RZ	0.24 ~ 0.6	6209-2RZ	3.72 ~ 9.3
6009-2RZ	2.2 ~ 5.5	6201-2RZ	0.32 ~ 0.8	6210-2RZ	4.4 ~ 11.0
6010-2RZ	2.4 ~ 6.0	6202-2RZ	0.4 ~ 1.0	6211-2RZ	5.8 ~ 14.6
6212-2RZ	7.4 ~ 18.4	6304-2RZ	1.2 ~ 3.0	6309-2RZ	7.6 ~ 19.0
6213-2RZ	8.4 ~ 20.6	6305-2RZ	2.04 ~ 5.1	6310-2RZ	10 ~ 25.1

(continued)

bearing model	filling grease amount	bearing model	filling grease amount	bearing model	filling grease amount
6300-2RZ	0.4 ~ 1.0	6306-2RZ	2.96 ~ 7.4	6311-2RZ	13 ~ 32.1
6301-2RZ	0.56 ~ 1.4	6307-2RZ	4.0 ~ 10.1		
6302-2RZ	0.72 ~ 1.8	6308-2RZ	5.5 ~ 13.8		
6303-2RZ	0.96 ~ 2.4				

① The contacting type sealing ball bearing filling grease amount is as the same as the non-contacting type sealing ball bearing filling grease amount.

There is some rules for the varied type bearing free space amount in a rolling bearing. As reference, according to the bearing bore diameter, the full filling grease amount in all free space of a bearing can be checked out from Figure 6-7. Every curve suitable for bearing type is listed in Table 6-14.

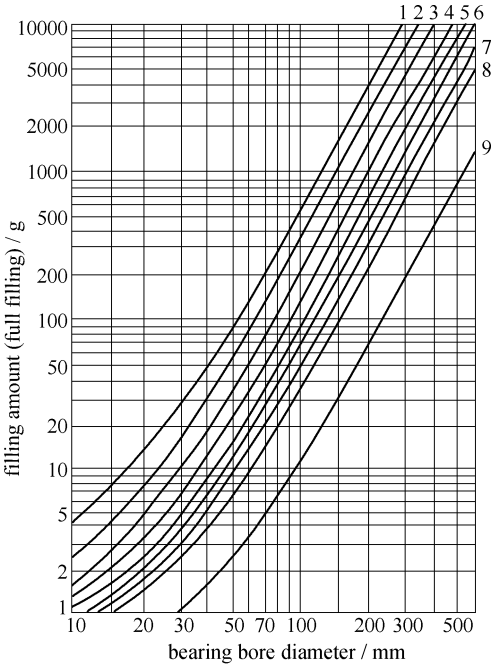


Figure 6-7 Every type, every serious bearing lubrication grease filling amount

Table 6 – 14 Every curve suitable for bearing type in Figure 6 – 7

singkrow deep groove ball bearing	curve	single row tapered roller bearing	curve	alighing roller bearing	curve	single row cylindrical roller bearing	curve
618TVHB	8 ~ 9	302 (7200)	3 ~ 4	213 (3300)	3	NU10 (32100)	7
618M	9	303 (7300)	2	222 (3500)	4	NU2 (32200)	5
160 (7000100)	7	313 (27300)	2	223 (3600)	2	NU22 (32500)	4
60 (100)	6	320X (2007100)	6	230 (3003100)	6	NU23 (32600)	2
62 (200)	4	322 (7500)	3 ~ 4	231 (3003700)	4	NU3 (32300)	3
63 (300)	2 ~ 3	323 (7600)	1 ~ 2	232 (3003200)	3 ~ 4	NU4 (32400)	2
64 (400)	1	329 (2007900)	7 ~ 8	239 (3003900)	8	NN30K (3182100)	5
角接触球轴承		330	5	240 (4003100)	5	NNU49 (4482900)	7
72B (66200)	4	332	4	241 (4003700)	3		
73B (66300)	2 ~ 3	331	4				
70 (46100)	6						

Note: the number in parentheses is the China old series symbol.

(2) Grease re-lubrication interval

If the lubrication grease life is lower than the bearing calculating life, the new grease should be filled in when the old grease still having good lubricating performance. Many factors such as the bearing type, the dimension, the rotation speed, the working temperature, the grease type and the environmental condition should be considered for determining the grease re-lubrication interval. Usually, it is very difficult to determine a bearing grease re-lubrication interval. It can be done to determine the grease re-

lubrication interval reasonably according to data from actual applications and tests and used experice.

1) The recommended method of determining the grease re-lubrication interval by SKF, which defined that the grease re-lubrication interval is a time period, at the end of which 99% of the bearings are still reliably lubricated, this represents the grease life.

The grease re-lubrication intervals t_f can be obtained from Figure 6 – 8 as a function of the speed factor A multiplied by the relevant factor b_f , where

$$A = n d_m$$

- n — rotational speed (r/min) ;
- d_m — bearing mean diameter $= 0.5 (d + D)$ (mm) ;
- b_f — bearing factor depending on bearing type and load condition ;
- C/P — load ratio.

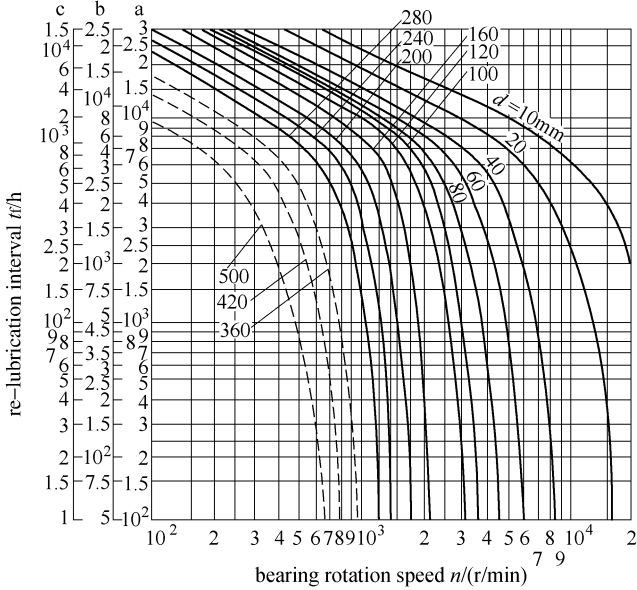


Figure 6 – 8 Recommended grease re-lubrication interval

- a: radio/ball bearing b: cylindrical roller bearing, needle roller bearing
- c: aligning roller bearing, Tapered roller bearing complement cylindrial roller bearing ($0.2T_f$). Crossed roller bearing with cage ($0.3T_f$) thrust roller bearing ($0.5T_f$)

The grease re-lubrication intervals t_i is an estimated value, valid for an operating temperature of 70℃, using good quality lithium thicker/mineral oil grease and is for bearings with rotating inner ring on horizontal shaft under normal and clean conditions. When bearing operating conditions differ, the grease re-lubrication intervals t_i should be adjusted. When using high performance greases, a longer re-lubrication interval and grease life may be possible.

2) Grease re-lubrication interval determining method by experience

Figure 6 – 9 is a grease re-lubrication interval determining method based on actual application experience data.

The value in Figure 6 – 9 is only suitable for the bearings of the stationary electric motor and the machine tool spindle. For other applications, this value must be modified, the grease re-lubrication interval is $q T_i$. Table 6 – 16 lists every typical equipment modifying q value.

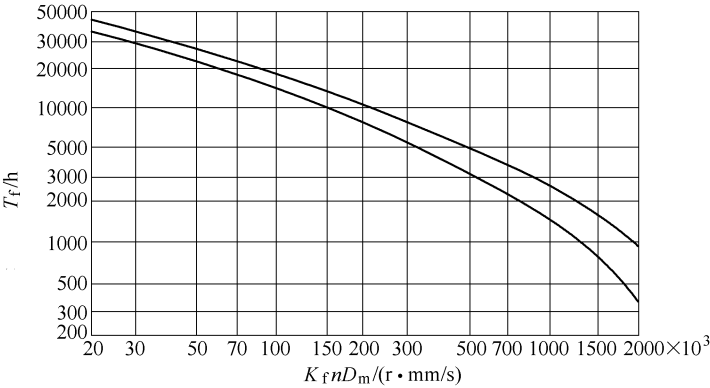


Figure 6 – 9 Grease re-lubrication interval experience curve

K_i — bearing structure type factor, checking out from Table 6 – 15

D_m — bearing mean diameter n — bearing rotational speed

Table 6 – 15 Bearing structure type factor K_i value

Bearing type	K_i
single row deep groove ball bearing	0.9 ~ 1.1
single row cylindrical roller bearing	1.8 ~ 2.3

(continued)

Bearing type	K_f
double row deep groove ball bearing	1.5
double row cylindrical roller bearing	2
single row angular contact ball bearing	1.6
full complement roller bearing without cage	25
double row angular contact ball bearing	2
thrust cylindrical roller bearing	90
spindle bearing contact angle	0.75
$\alpha = 15^\circ$	0.9
$\alpha = 25^\circ$	
needle roller bearing	3.5
tapered roller bearing	4
four-point contact angular contact ball bearing	1.6
single row aligning roller bearing	10
aligning ball bearing	1.3 ~ 1.6
aligning roller bearing without middle rib (E type)	7 ~ 9
thrust ball bearing	5 ~ 6
double row thrust angular contact ball bearing	1.4
aligning roller bearing with middle rib	9 ~ 12

Table 6 – 16 Every typical equipment q value

application	dust, moisture	shocks, vibrations	working temperature	load value	air flow condition	reduction factor q
stationary electric motor	—	—	—	—	—	1
lathe spindle	—	—	—	—	—	1
grinding spindle	—	—	—	—	—	1
face grinder	—	—	—	—	—	1
circular saw shaft	●	—	—	—	—	0.8

(continued)

application	dust , moisture	shocks , vibrations	working temperature	load value	air flow condition	reduction factor q
flywheel of a car body press	●	—	—	—	—	0.8
hammer mill	●	—	—	—	—	0.8
dynamo-meter	—	—	●	—	—	0.7
journal roller bearing of locomotive	●	●	—	—	—	0.7
electric wind machine	—	—	—	—	●	0.6
rope , sliding wheel	● ●	—	—	—	—	0.6
car front shaft	●	●	—	—	—	0.6
textile spindle	—	● ● ●	—	—	—	0.3
jaw type crusher	● ●	● ●	—	●	—	0.2
vibration motor	●	● ● ●	●	—	—	0.2
wire section roll (paper making machine)	● ● ●	—	—	—	—	0.2
paper making machine roll	● ● ●	—	●	—	—	0.2
steel mill work roll	●	—	—	● ●	—	0.2
centrifugal machine	●	—	—	●	—	0.2
excavator	● ● ●	—	—	●	—	0.1
saw machine	●	● ● ●	—	—	—	0.1
shock , roll wheel	●	● ● ●	● ● ●	—	—	< 0.1
vibration screen	●	● ● ●	—	—	—	< 0.1

(continued)

application	dust , moisture	shocks , vibrations	working temperature	load value	air flow condition	reduction factor q
shoveling machine	● ●	—	—	● ● ●	—	< 0. 1
reel printing machine	●	—	●	● ● ●	—	< 0. 1
leather belt transmission roll	● ● ●	—	—	●	—	< 0. 1

note: ● — normal; ● ● — moderate to serious; ● ● ● — very serious.

3. Lubrication grease performance test

For some very important equipments, in order to properly selecting lubrication grease, it is always required to carry the grease performance test. Many countries have their own relevant conventional grease performance test standard, such as ASTM in USA, IP in England, JISK in Japan, DIN in Germany. Except for the conventional grease performance test, many grease application tests have been more actual significance for grease selection. Therefore, the bearing using grease life test has been paying more attention.

Table 6 – 17 lists grease performances, norm items, standards and test methods; Table 6 – 18 lists rolling bearing using grease application tests, Table 6 – 19 lists grease life test methods.

Table 6 – 17 Grease performances, norm items, standards and test methods

quality	item	norm	test method	test brief, test condition, summary
color	appearance	light yellow -brown	witness	including the color, the transparent degree, the adhering feature, the homogeneous and the fiber condition
consistence	penetration degree (25℃ , 150g) / (1/ 10mm)	220 ~ 250	GB/T 269	using penetration degree meter, on 25℃ , measuring the sinking depth in the measured grease by a 150 g standard taper body, the value is calculated in 1/10mm, the value is more larger that means the grease is more soft. On the contrary, means the grease is hard

(continued)

quality	item	norm	test method	test brief, test condition, summary
heat resistance performance	dropping point/℃ (not lower than)	100	GB/T4929 SH/T 0115	using grease dropping point meter, measuring the dropping temperature after the measured grease dissolving. The using temperature can be determined based on the dropping point. Except for special condition, the grease working temperature is 0.7 times the dropping point, but the base oil flash point should also be considered
	evaporating amount (%) 100℃ , 22h (not larger than)	2	GB/T 7325	On the stipulating temperature and time, measuring the evaporating loss amount, the smaller, the better
	high temperature penetration degree (150℃ , 3h) (not larger than)	300		using penetration degree meter (non-work), on the stipulating temperature and time, measuring its penetration degree value, the smaller, the better, if the value is too large, it is easy to flow out from the bearing, so, it is can not be used
	steel net dividing oil (100℃ , 3h) (not larger than)	3	SH/T 0324	Putting the grease into a steel net, on the stipulating temperature and time, measuring the dividing oil amount, the smaller, the better, if it is larger, it is expressed that this grease can not be used in this temperature
	leaking amount (%) (not larger than)	5	SH/T 0326	Putting certain sample in a wheel bearing test machine, running on the stipulating temperature and time, measuring the leaking amount for checking the grease using condition, temperature and the adhesive force, the smaller, the better, if it is larger, this grease can not be used in this temperature

(continued)

quality	item	norm	test method	test brief, test condition, summary
water resisting performance	water pouring (38℃ , 1h)	under 10	SH /T0109	Putting certain sample into water pouring tester, after spraying 38℃ water 1h in 5mL per min, measuring the loss amount, the smaller, expressing the better water resistant; the contrary, the bad
	cylinder roll test adds water 10% (not larger than)	300	SH /T122	adds 10% water into grease, on the stipulating time running and mixing, measuring the grease changing by miniature penetration degree converting as standard penetration degree, $1/4$ miniature penetration degree $A = 3.75B + 24$
	add water stability (adds water 10% 100000 times) (not larger than)	400	GB/T 269	adds 10% water into grease, after mixing, pounding 100000 times by the pounding grease tester, measuring the penetration degree changing, the more smaller changing, expressing the more better water resistance performance. On the contrary, the bad.
machinery stability	Shearing stability (100000 times) (not larger than)	350	GB/T 269	pounding 100000 times by the pounding grease tester, measuring the penetration degree, the more smaller, the better machinery stability, On the contrary, the bad
	strengthen limit /℃ (not smaller than)		SH /T0323	under stipulating temperature, measuring the maximum pressure when the grease moving in the plasticity meter thread tube, expressing by Pa

(continued)

quality	item	norm	test method	test brief, test condition, summary
pressure resistance performance	four ball tester PB value (not smaller than)		SH /T0202	PB or PK value is the same. The maximum testing blocking load is expressed by N, the sintering load is expressed by N
	Timken test OK value (not smaller than)		SH /T0203	OK value: the testing bruise load weight expressed by lbf (or convert as N)
oxide stability	oxide stability (expressed by the pressure difference or the acid value)		SH /T0325 SH/T0335	using 20 g grease in oxygen pellet, under 784 k Pa oxygen pressure, oxidizing 100℃ , 100h (or 500h), measuring the pressure reducing value or measuring the free alkali or the free organic acid changing
low temperature flowing ability	low temperature running torque	starting torque	SH/T 0338	inspecting the starting torque and the running torque value for the grease, for determining the temperature performance condition
	resemble viscosity℃ D = 10 (1/ s) (Pa. S) (not larger than)	running toque	SH/T0048	inspecting the grease flow ability under the stipulating temperature, specially for the concentrating supply oil, this norm is very important, calculating in pressure, speed, capillary thickness according to equation, in Pa. S
machinery impurity	acid dividing method, impurity (%) (not larger than)	GB/T513		filtering the acid resolving greasesample, calculating the left impurity mainly is the wearing impurity, usually the stipulation is “no”
	extracting method (%) (not larger than)		GB/T511	extracting 10g grease sample by solvent, measuring the not dissolving in the solvent and the hot distilled water impurity, the permissible amount in the grease is not more than 5%

(continued)

quality	item	norm	test method	test brief, test condition, summary
machinery impurity	inspecting harmful particle		SH/T0322	inspecting and estimating the harmful particle number in the grease by magnifying glass and judging the grade
	microscope inspecting impurity	over 25 μm over 75 μm over 125 μm	SH/T 0336	inspecting and estimating the harmful impurity number by microscope, in number/ cm^3
antiseptic anti-rust feature	corroding test (copper piece , steel piece, iron cast piece)	qualified	SH/T0331	using copper piece, steel piece or iron cast piece based on the machine structure, usually, the stipulation are 100 $^{\circ}\text{C}$ and 3h, then taking out the t test piece and observing the metal piece color changing, inspecting the corrosion degree
	bearing anti-rust 52 $^{\circ}\text{C}$, 48h (relative humidity100%)	X grade	GB/T5018	filling grease sample into bearing, under relative humidity 100% , 52 $^{\circ}\text{C}$, running 48h, inspecting the bearing if there is pit, score, rust, or black spot. The first grade is no any, the second grade is 2, the third grade is more
	wet heating steel piece, 30h	qualified	GB/T2361	smearing grease sample on the steel piece, under relative humidity 95% , 47 $^{\circ}\text{C}$, stay 30h, inspecting if there is corrode
	salt mist test (steel piece)	qualified	SH/T0081	smearing grease sample on the steel piece, under 35 $^{\circ}\text{C}$ temperature, spraying 5% salt water for stipulation time, inspecting if there is corrode
	protecting performance 50 $^{\circ}\text{C}$, 30h	qualified	SH/T0333	putting the steel piece into grease sample, keeping 30h on 50 $^{\circ}\text{C}$, inspecting if there is corrode

(continued)

quality	item	norm	test method	test brief, test condition, summary
dividing oil	pressure dividing oil (not larger than)		GB/T392	measuring the grease dividing oil amount on the stipulating temperature, time and weight, measuring the grease dividing oil amount. Mainly inspecting the dividing oil condition during the storing period
life	bearing life		ASTM D1741	putting the grease sample into the test bearing, running on 125℃ condition, if the temperature is over 10℃, the noise increasing longer than 10 min, judging as bearing damaged, the running time in h
hardening	surface hardening test (50℃, 24h) miniature penetration degree difference (not more than)		another stipulation	putting the grease into a grease cup, measuring the miniature penetration degree without working, then put in 50℃ constant temperature box, keeping for 24h, then cooling to 25℃, measuring the miniature penetration degree, calculating the difference
water content	water content (%) (not more than)		GB/T512	measuring the water content in the grease by grease water content meter, the smaller, the better

Table 6 – 18 Bearing grease application test

test item	test summery	test result evaluation
grease centrifugal dividing oil	the grease test tube with 2g grease on 1300r/min running, measuring the dividing oil weight	can be sensitively and quickly determine the grease dividing oil weight and determine the grease storing time period
grease oven test	put the grease into the oven, observing the changing condition under 150℃	inspecting the oxide resistance and high temperature resistance performance

(continued)

test item	test summery	test result evaluation
grease Fafner test	after tester running 50h , evaluating the thrust bearing weight changing	test grease resisting micro moving wear performance. Used in railway vehicle
grease EMCOR test	running in EMCOR tester for 108 h	evaluating grease resisting water and resisting corrosion ability
grease noise test	measuring bearing with grease noise in anechoic chamber by sound meter	analyzing the bearing grease noise grade and reason
grease vibration test	measuring bearing vibration level in ANDERON vibration tester (in CHINA is BVT-1 bearing vibration tester) , under 1800r/min	analyzing the bearing vibration level and reason
grease temperature rising test	recording 2h bearing outer ring temperature changing on Fed 791-331 tester according to the stipulation norm	evaluation the grease temperature rise
bearing grease leaking test	after 22h running on ASTM 1741-64 tester measuring the grease leaking amount	evaluating the bearing grease leaking amount, inspecting the sealing ability

Note: In China there is the bearing dust proof, grease leaking, and grease temperature rise standard JB/T 8571 — 1997.

Table 6 – 19 Grease life test method

test standard	bearing model	summary
ASTM 1741	2 sets 6306 deep groove ball bearing grease $6 \pm 0.1\text{g}$	rotational speed: 3500r/min, load: radial 110N, axial 176N, temperature 125℃, tester running 20h, stop 4h until the bearing damaged
ASTM 3527	LM 67048/67010 LM 11949/11910	rotational speed: 1000r/min, temperature 150℃, imitating car wheel bearing

(continued)

test standard	bearing model	summary
Fed 791B 331. 2	One 6204 deep groove ball bearing grease 3 ±0. 1g	rotational speed: 10000r/min, temperature 176℃, load: radial 14N, axial: 23N, running 21. 5h, stop 2. 5h until the bearing damaged
Fed 791B 331. 1	One 6204 deep groove ball bearing grease 3. 2 ±0. 2g	rotational speed: 6000r/min, temperature 232℃, load: radial 68N, axial: 23N, running 20h, stop 4h until the bearing damaged
FAG 18	7206	rotational speed: 6000r/min, axial load: 1. 5kN, temperature 120℃
DIN 51806	22312 aligning roller bearing 2 sets filling 50 g grease	rotational speed: 1500r/min or 2500r/min, load: radial 8. 5kN, temperature 125℃, running 6 h to 20 days
IP-168	BRM040 deep groove ball bearing one set, filling grease 28g ± 0. 1g	rotational speed: 10000r/min, temperature 200℃, radial load 227 ~ 1361N, running 50h

6. 3. 4 Oil Lubrication

The lubrication oil selection mainly is based on the lubrication oil viscosity to guarantee forming a certain thickness oil film, preventing the metal to metal directly contacting under the bearing working temperature. The lubrication oil should be with the minimum necessary viscosity value.

Usually, the rolling bearing lubrication is using the mineral oil without additive, the minimum necessary viscosity ν_1 under the bearing working temperature can be checked out from Figure 6 - 10. The lubrication oil parameter can be expressed by the viscosity ratio K .

$$K = \frac{\nu}{\nu_1}$$

Where ν — lubrication oil viscosity under the bearing working temperature;
 ν_1 — the minimum necessary viscosity under the bearing working temperature.

The lubrication oil viscosity is related to the temperature, the viscosity index is expressed by *VI*. The recommended viscosity index is at least 85 while selecting lubrication oil.

Figure 6 – 11 shows the viscosity index with the temperature diagram. This diagram is suitable for the ISO listed lubrication oil viscosity index grade and the viscosity range in 40°C .

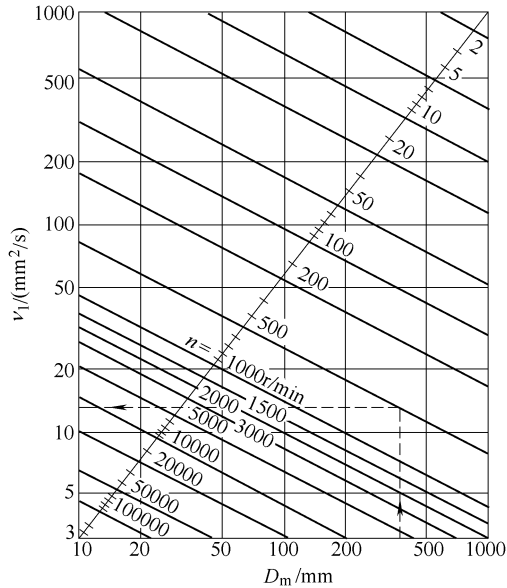


Figure 6 – 10 Minimum necessary viscosity ν_1

Table 6 – 20 ISO viscosity grade and kinetic viscosity

ISO viscosity grade	viscosity range in 40°C/ (mm ² /s)		
	mean	minimum	maximum
ISO VG 2	2. 2	1. 98	2. 42
ISO VG 3	3. 2	2. 88	3. 52
ISO VG 5	4. 6	4. 14	5. 06
ISO VG 7	6. 8	6. 12	7. 48
ISO VG 10	10	9. 00	11. 0
ISO VG 15	15	13. 5	16. 5

(continued)

ISO viscosity grade	viscosity range in 40°C/ (mm ² /s)		
	mean	minimum	maximum
ISO VG 22	22	19. 8	24. 2
ISO VG 32	32	28. 8	35. 2
ISO VG 46	46	41. 4	50. 6
ISO VG 68	68	61. 2	74. 8
ISO VG 100	100	90. 0	110
ISO VG 150	150	135	165
ISO VG 220	220	198	242
ISO VG 320	320	288	352
ISO VG 460	460	414	506
ISO VG 680	680	612	748
ISO VG 1000	1000	900	1100
ISO VG 1500	1500	1350	1650

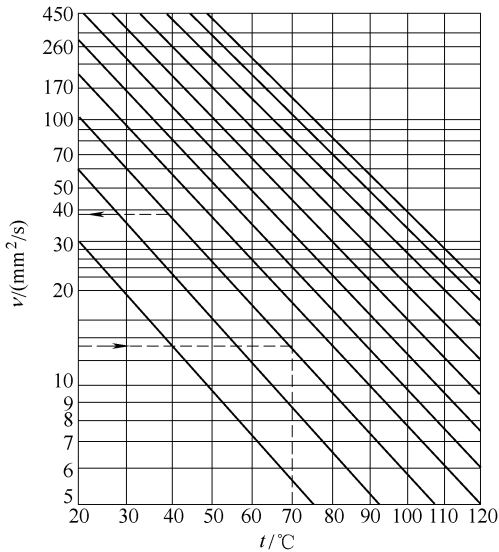


Figure 6 – 11 Mineral lubricating oil viscosity — temperature diagram

When selecting lubrication oil, first of all, according to the bearing mean diameter and the rotational speed, the minimum necessary viscosity ν_1 value can be checked out from Figure 6-10, and then according to the minimum necessary viscosity ν_1 value and the bearing working temperature, the reference 40°C viscosity value can be checked out.

[**Example 6 - 1**] Bearing bore diameter $d = 340\text{mm}$, outer diameter $D = 420\text{mm}$, bearing rotational speed $n = 500\text{r/min}$, bearing running temperature is 70°C. From Figure 6-10, the minimum necessary viscosity ν_1 value under working temperature is $13\text{mm}^2/\text{s}$, and then, the reference 40°C viscosity value $39\text{mm}^2/\text{s}$ can be checked out based on $D_m = 0.5(d + D) = 380\text{mm}$ from Figure 6-11. Thus, it should be selected that the reference 40°C viscosity value is larger than $39\text{mm}^2/\text{s}$.

If the viscosity ratio $K = \nu/\nu_1$ is smaller than 1, the lubrication oil with EP additive should be selected (if the viscosity ratio $K = \nu/\nu_1$ is smaller than 0.4, the lubrication oil with EP additive must be selected). If K is larger than 1, the lubrication oil with EP additive could raise the bearing running reliability.

Table 6-21 lists the commonly used lubrication oil performance in China. Table 6-22 lists the machine tool rolling bearing commonly used lubrication oil quality norm.

As reference, Table 6-23 lists some bearing application mean temperature summary values.

2. Lubrication method selection

The rolling bearing lubrication method selection is related to many factors, such as the bearing rotational speed, the load value, the temperature rise and the bearing type. The commonly used lubrication method are as follows:

(1) Dripping oil lubrication

As shown in Figure 6-12, the lubrication oil is dripping in the required bearing position lubricated by oil self weight through the needle valve type oil cup. For the vertical shaft, the bearing axis is vertical, it is easy to enter in to the bearing by oil self weight for oil. When the shaft is in

Table 6-21 Commonly used lubrication oil performance in China

name	trademark	using temperature range℃	viscosity/ (mm^2/s) (50℃)	viscosity feature	heat stability heat oxide	mutual protect with metal and rubber	suitable		
							load	temp. range	shearing
low temp meter oil	s-3	-60 ~ +120	11 ~ 14	mid		OK	small	mid-low	small
low temp. meter oil	s-4	-60 ~ +120	11 ~ 14	better		OK	small	mid-low	small
low temp meter oil	s-5	-60 ~	18 ~ 23	mid.		OK	small	mid-low	small
low temp meter oil	s-14	-60 ~ +120	22.5 ~ 28.5	mid.		OK	large	mid-low	large
low temp meter oil	s-16	-60 ~	19 ~ 25	mid.		OK	mid	highbetter	mid
high temp. meter oil	4112	-60 ~ +120 short time to +150	50℃ not smaller than 8 20℃ not smaller than 20-50℃ not smaller than 8000	good	good	bad for rubber good for metal	small	mid-high	large

(continued)

name	trademark	using temperature range℃	viscosity/ (mm ² /s) (50℃)	viscosity feature	heat stability heat oxide	mutual protect with metal and rubber	suitable		
							load	temp. range	shearing
high temp meter oil	4113	−60 ~ +120 short time to +150	50℃ 11 ~ 14 −50℃ not larger than 8000	good	good	OK	mid	mid-high	mid
high temp meter oil	4114	−60 ~ +120 short time to +150	50℃ 18 ~ 23 −50℃ not larger than 8000	good	good	OK	mid	mid-high	mid
high temp meter oil	4115	−60 ~ +120 short time to +150	50℃ 18 ~ 23 −50℃ not larger than 8000	good	good	OK	mid	mid-high	mid
high temp meter oil	4116	−60 ~ +150	50℃ not over 36 , −50℃ not larger than 8000	better	better	OK	mid	low-high	small
precise machine tool oil	10	0 ~ 100	50℃ 8 ~ 13				large	mid	mid-large

Table 6 - 22 Machine tool rolling bearing common used lubrication oil quality norm

name and trademark		kinetic viscosity (40℃)	kinetic viscosity (50℃)	solidifying point ≤ (%)	remain charcoal ≤ (%)	acid value ≤ mgKOH/s	machinery impurity ≤ (%)	water content ≤ (%)	flash point ≤ /℃	color No. not notlarger
spindle oil	N2	2.0 ~ 2.4	1.7 ~ 2.0	- 15			no	no	60	test
	N3	2.9 ~ 3.5	2.4 ~ 2.9						70	
	N5	4.2 ~ 5.1	3.3 ~ 4.0						80	
	N7	6.2 ~ 7.5	4.8 ~ 5.7						90	
	N10	9.0 ~ 11.0	6.8 ~ 8.1						100	
	N15	13.5 ~ 16.5	9.8 ~ 11.8						110	
	N22	19.8 ~ 24.2	13.9 ~ 16.6						120	
machine oil	N5	4.14 ~ 5.06		- 10		0.04	no	no	110	8
	N7	6.12 ~ 7.48		- 10		0.04	no		110	8
	N10	9.00 ~ 11.00		- 10		0.04	no		125	8
	N15	13.5 ~ 16.5		- 15	0.15	0.14	0.005		165	9
	N22	19.8 ~ 24.2		- 15	0.15	0.14	0.005		170	13
	N32	28.8 ~ 35.2		- 15	0.15	0.16	0.005		170	15
	N46	41.4 ~ 50.6		- 10	0.25	0.2	0.007		180	20
	N68	61.2 ~ 24.2		- 10	0.25	0.35	0.007		190	20
steam turbine	HU-20		18 ~ 22	- 15		0.03	no		180	
	HU-30		28 ~ 32	- 10		0.03			180	

**Table 6 – 23 1 Some bearing application mean
temperature summary value**

supporter type	working temperature /°C	supporter type	working temperature /°C
planer spindle	40	wire milling roll supporter	65
desk-type drilling machine	40	vibration motor	70
horizontal drilling machine	40	wiring machine	70
circular saw	40	vibration screen	80
steel billet and plate billet	45	ship propeller thrust supporter	80
lathe spindle	50	vibration mill roll	90
vertical boring machine	50	hammer breaker	80
double bracket frame saw	50	traction motor	80 ~ 90
wood milling spindle	50	paper making machine dry roll	120 ~ 130
paper making machine press bright roll	55	heat air fan	≈ 90
heat milling belt mill bracket supporting roll supporter	55	car engine pump	≈ 120
		turbine air compressor	≈ 120
surface grinding machine	55	internal-combustion engine crankshaft	≈ 120
jaw type breaker	60	plasticity calender	≈ 180
railway vehicle box supporter	60	oven wheel hub bearing	200 ≈ 300
hammer grading machine	60		

the horizontal position, the oil can be dripped firstly on the shaft end nut, and then splashing to the bearing. The dripping oil amount can be adjusted, usually just 5 ~ 6 drops/min. This lubrication method is simple and convenient, which is used in small type bearing at the high speed running applications.

(2) Bath (sump) lubrication

The bath lubrication method is suitable for low speed and moderate

speed applications. A part of bearing is intruding in the oil, as shown in Figure 6 – 13. While the bearing is running, part of the bearing is immersed in oil. During each revolution of the cage, the rolling elements are freshly dipped into the oil once and then with oil passed on the raceway and other sliding working surfaces. When the shaft is in the horizontal position, the oil level should be such that the lowest rolling element center position is covered when the bearing is not rotating. At a higher oil level, churning of the oil in the housing would be raising the temperature, accompanied by foaming and accelerated the oil aging. Only at low speed rotating, the higher oil level is permissible with little churning. In order to observing the oil level, an oil lever indicator or dipstick should be provided at the housing.

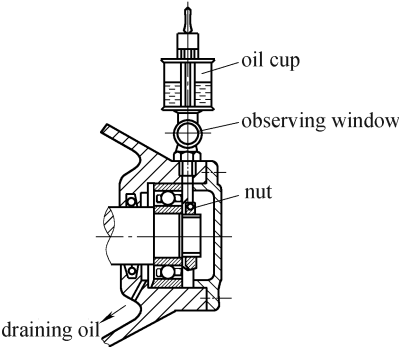


Figure 6 – 12 Dripping oil lubrication

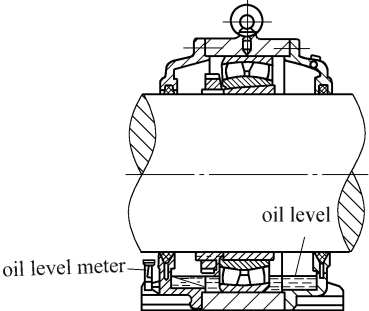


Figure 6 – 13 Bath oil lubrication

(3) Splashing oil lubrication

As shown in Figure 6 – 14, the bearing is lubricated by using the gear rotating or the oil flying ring rotating. When the bearing is located in the position where the oil is not easy arriving, a guiding oil groove or a guiding oil slot should be provided on the bearing housing. This lubrication

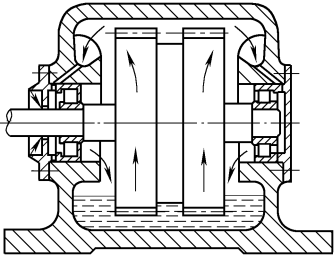


Figure 6 – 14 Splashing oil lubricating

method is simple, but the starting condition is not well, which is used in close housing where the splashing oil easy arriving.

(4) Circulating oil lubrication

The circulating oil lubrication method is mainly used in the higher speed and heavier load applications. The filtering lubrication oil is pumped to the bearing position by hydraulic pump (pressure is about 0.15MPa) and then returning to the oil box after throughout the bearing, and through re-filtering, cooling, then reusing again. Figure 6-15 shows the three kinds of the circulating oil lubrication examples.

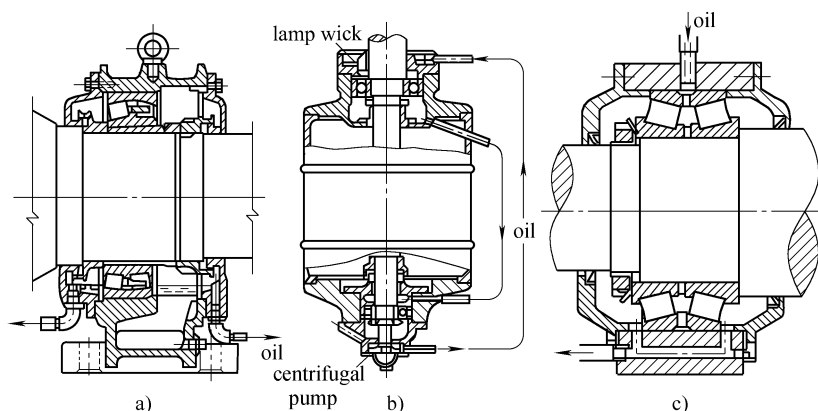


Figure 6-15 Circulating oil lubrication

(5) Spraying lubrication

When the bearing rotates in high speed, it is very difficult to transport the lubrication oil into the bearing because the rolling elements and the cage also rotate in the appropriate high speed forming air flowing by the surrounding air. In this case, the lubrication oil has to be sprayed into the bearing. Its working principle is that the lubrication oil is sprayed into the bearing through one or many spray nozzles between the inner ring and the cage by the hydraulic pump. After through the bearing, the lubrication oil flows into the oil slot, at the same time the bearing can also be cooled, as shown in Figure 6-16.

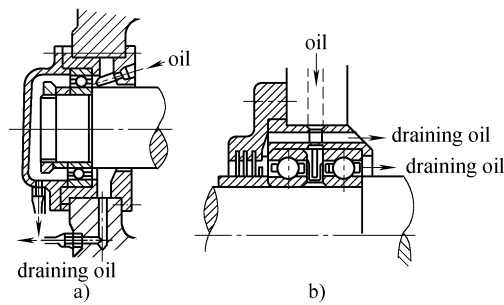


Figure 6-16 Spraying oil lubrication

Usually, the nozzle diameter is as $0.7 \sim 2\text{mm}$. The nozzle diameter can not be too small, otherwise the nozzle would be blocked. In order to prevent the nozzle from blocking, usually the filter is provided.

Figure 6-17 is a spraying lubricating device using spraying ring. There is a spraying lubricating ring 1 between two bearings for transporting the lubricating oil into the bearing through the nozzles 3 on the ring circumference. The slot 2 of the spraying ring links all nozzles. The oil flows out from the bearing through the notch 4 of the spraying ring. In order to prevent the nozzle from blocking, the oil outlet must be large enough, on some times, it is required to pump the oil out.

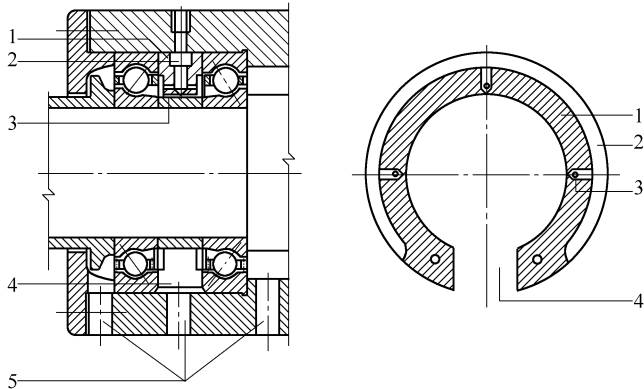


Figure 6-17 Spraying oil lubrication with spraying ring

1 — spraying ring 2 — linking slot 3 — nozzle 4 — oil outlet notch 5 — oil outlet

The spraying lubrication oil pressure is as 0.1 ~ 0.5MPa, the oil flow rate should be determined by removing heat amount from the oil and that is related to the bearing dimension. Table 6 – 24 lists the relationship between the spraying lubrication oil amount and the bearing bore diameter.

The spraying lubrication nozzle diameter and the spraying pressure can be selected based on Figure 6 – 18.

Table 6 – 24 Relationship between the spraying lubrication oil amount and the bearing bore diameter

Bearing bore diameter/mm	Oil amount (L/min)
≤ 50	0.5 ~ 1.5
> 50 ~ 120	1.1 ~ 4.2
> 120	2.5

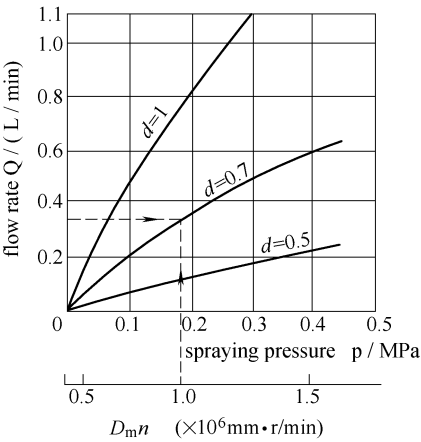


Figure 6 – 18 Spraying nozzle diameter and spraying pressure

(6) Oil mist lubrication

The oil is misting from compressed air and spraying into the bearing.

The air pressure is as 0.05 ~ 0.15MPa, the air amount for every bearing is as 5 ~ 10L/min, the supply oil amount is as 0.1 ~ 1mL/min. Because of the compressed air and the oil mist together entering into the bearing, the cooling effect is better. The oil mist lubrication is mainly

suitable for the high speed bearings. Figure 6 – 19 is an oil mist lubrication example.

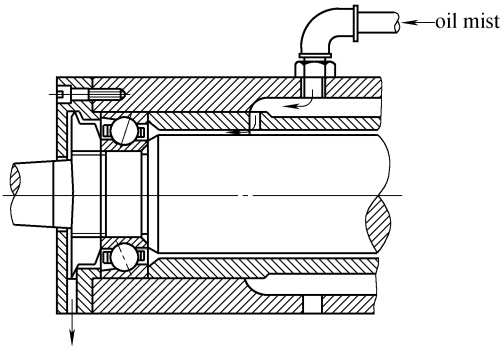


Figure 6 – 19 oil mist lubrication

The using oil amount of oil mist lubrication is very small. Table 6 – 25 lists the relationship between the supply oil amount and the bearing bore diameter.

The oil misting generator has been specialized production and many specifications can be chosen according to the supply oil amount, the using air amount, the compressed air spraying nozzle diameter and the guiding tube diameter etc.

The oil viscosity used in the oil mist lubrication should be lower than $200\text{mm}^2/\text{s}$, the oil mist should be formed.

Table 6 – 25 Relationship between the supply lubrication oil amount and the bearing bore diameter

bearing bore diameter/mm	oil amount (mL/h)
~ 50	0. 1 ~ 0. 2
> 50 ~ 120	0. 2 ~ 0. 5
> 120	0. 5 ~ 1. 0

(7) Oil-air lubrication

The quantitative column pump distributor pumps periodically (1 ~ 60 mins) the little amount lubrication oil (0. 01mL ~ 0. 06mL) in fixed

quantity for mixing with the compressed air (pressure 0.3 ~ 0.5 MPa, flow rate 20 ~ 50 L/min) and then spraying into the bearing through the nylon pipe (the inner diameter is about 2 ~ 4 mm) and the nozzle mounted on the bearing adjacent position per certain time interval.

The main difference of the oil-air lubrication with the oil mist lubrication is that the supply lubrication oil is not misting, but is in drop form entering to the bearing. Therefore, it is much easier to deposit than the mist lubrication, and is not contaminating the environment. Because a large amount air is used in order to cooling the bearing, the bearing running temperature is lower than other lubrication methods. So the oil-air lubrication method is specially suitable for the high speed bearing lubrication, suitable for $D_m n > 10^6 \text{ mm} \cdot \text{r/min}$ applications.

Figure 6-20 is oil-air lubricating principle sketch. In this sketch, the single line expresses the compressed air pipe line, the double line expresses the lubrication oil tube line. The time relay 9 controls the electromagnetism valve in the fixed time to let the compressed air entering into the air-hydraulic pump. The lubrication oil is pumping out from the oil box 3 and is transporting to the quantitative column pump distributor 5 from the air-hydraulic pump. After the single direction valve, the oil-air mixture is introduced at the periphery by way of distribution ring containing a number of nozzles. The pressure relays 4 and 8 are used for controlling the oil and the air pressures, the throttle 7 is used for controlling the spraying air pressure.

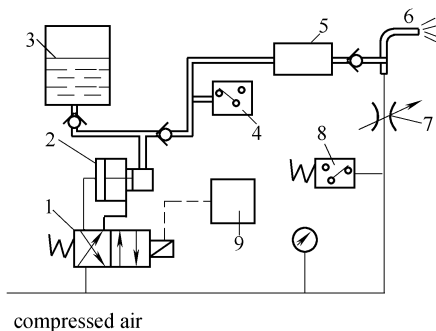
The using lubricating oil viscosity in the oil-air lubricating method is usually as $10 \sim 40 \text{ mm}^2/\text{s}$.

In order to reducing the supplying oil non-uniformity, it is proper to reduce the draining oil amount and the draining time interval per times. The commonly used draining oil amount is as 0.01 ~ 0.03 mL, and the relevant draining time interval is as 1 ~ 16 min.

For high speed rotating bearing, in order to spray reliably the lubrication oil into the bearing, the more attention should be paid to the spraying nozzle shape and the mounting position. Usually, the nozzle should

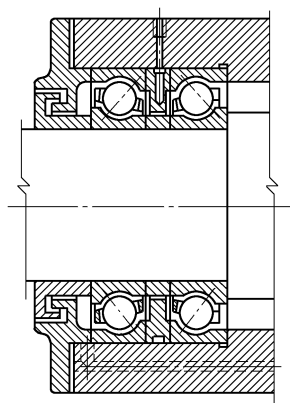
be mounted on every bearing position and the spraying oil amount into every bearing can be adjusted. The nozzle spraying hole diameter is as $0.5 \sim 1\text{ mm}$. The spraying nozzle should be mounted on the position that is between the inner ring and the cage and the direction should be towards the contact point of the inner ring raceway and the rolling element.

Figure 6-21 is a real using example of the oil-air lubrication method, an air-oil ring with a number of nozzles is mounted on the middle position of two bearings, the oil-air mixture is transporting into the bearing through the oil-air ring nozzle.



**Figure 6-20 Oil -air lubrication
principle sketch**

- 1 — electromagnetism valve 2 — pump
3 — oil box 4, 8 — pressure relay
5 — quantitative column pump distributor
6 — spraying nozzle 7 — throttle 9 — time relay



**Figure 6-21 Oil-air
lubrication**

(8) Centrifugal lubrication

For the vertical shaft support, the lubrication oil can be transported to the upper part from the lower part by using the tapered hole pumping action. Figure 6-22, Figure 6-23 and Figure 6-24 are the typical examples.

Figure 6-22 is the actual using example, in which the lubrication oil is transported into the circular slot on the middle part from the lower part by

using the pumping action of the tapered shape crack between the vertical shaft tapered shaft end and the housing and then raising to the upper part bearing position through the pipe line.

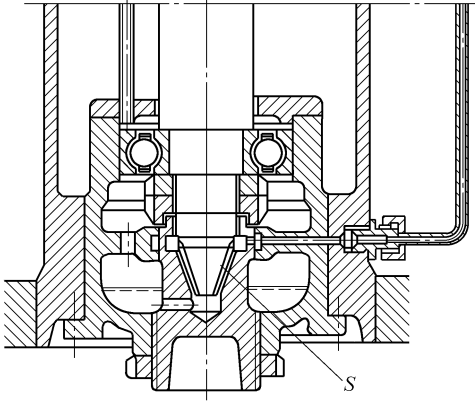


Figure 6 - 22 Spindle tapered shape crack transporting oil

Figure 6 - 23 is the actual using example, in which the lubrication oil is transported into the upper part by a tapered shape sleeve that is pressed into the high speed shaft lower end and into the upper bearing through the three radial holes on the upper part.

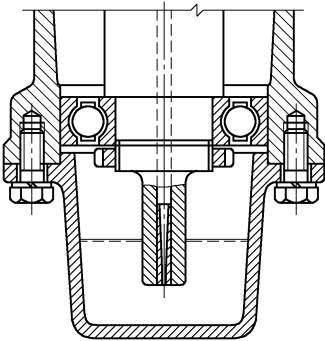
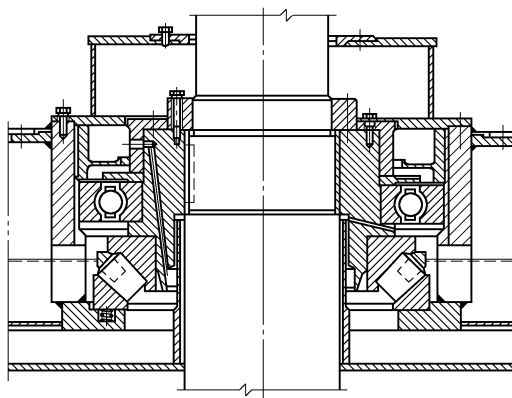


Figure 6 - 23 Shaft end tapered shape sleeve transporting oil

Figure 6 - 24 is the actual using example, in which the lubrication oil is transported into the upper part deep groove ball bearing through the inclined hole on the shaft, the lower part of the rotor is immersing into the oil pool. On the above part of the deep groove ball bearing, the oil is flowing to a fixed oil plate type collector through the radial hole, and then dropping into the bearing.

The centrifugal lubrication method should be determined based on the concreting condition though testing. There are many factors related to the

centrifugal lubrication, such as the transporting oil amount and its viscosity, the transporting oil gap width, the entering oil pipe line diameter and its length, the tapered angle of the transporting oil component and the rotational speed. Usually the transporting oil amount is quite little, in order to keeping the reliably lubrication within the time interval of the starting stage to the lubricating oil arriving into the bearing, a little amount of lubrication oil should be keeping in the bearing.



**Figure 6-24 Shaft end tapered shape transporting oil
and inclined hole transporting oil**

(9) Concentration lubrication

When many machines are rotating at the same time, there are many bearings requiring lubrication. In this case the concentration lubrication method can be used, as shown in Figure 6-25. The concentration lubrication is being driven by the motor, the pump 1 is intermittently switched by the control device 6. The metering valve 3 allows the mechanism to adapt the lubrication oil quantities to the requirements of each individual bearing. The lubrication oil can reach the remote positions through the longer lubrication pipes. The concentration lubrication system offers the further advantage of an easy supervision of pump and central reservoir. The concentration lubrication system can also be operated with soft greases.

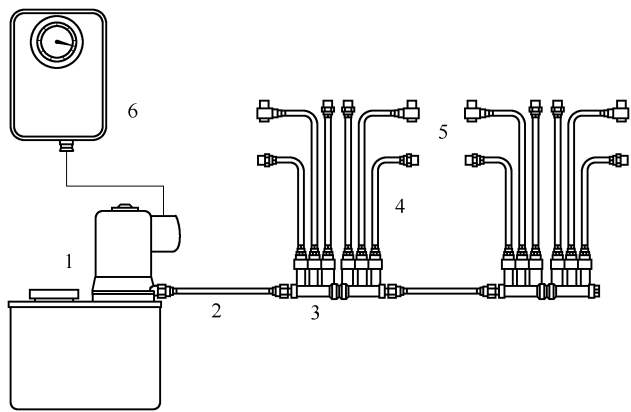


Figure 6 – 25 Concentration lubrication system

- 1 — hydraulic pump 2 — main oil pipe line 3 — metering oil valve
4 — dividing oil line 5 — lubricating position 6 — control device

3. Re-lubrication interval

The lubrication oil change interval is related to the running condition and the oil amount. When the bath lubricating method is used, if the running temperature is not over 50℃ and without contaminating, usually it is once a year to change the oil. The much higher the temperature is, the much times the changing oil should be. If the running temperature is close to 100℃ , the changing oil interval must be once per three months.

When the circulating oil lubrication method is used, the changing oil interval should be determined according to the circulating oil rate and if the machine oil is or is not contaminating. The proper changing oil interval only can be determined through the relevant testing and analyzing the lubricating condition. Table 6 – 26 is N32, N46 machine oil changing oil interval.

Table 6 – 26 N32, N46 machine oil changing oil interval norm

Item	Changing oil interval norm
kinematic viscosity (40℃) changing rate (%)	over ±15
acid value (mg KOH/g)	>0. 5
color value	larger than new oil 3 number

(continued)

Item	Changing oil interval norm
machinery impurity	> 0. 2
iron including (ppm)	> 100
water content	> 0. 1

Note: The kinematic viscosity (40℃) changing rate (%) is calculated based on the following equation:

$$\chi = (\nu_2 - \nu_1) / \nu_1 100\%$$

where ν_1 — new oil viscosity standard central value (mm²/s) ;
 ν_2 — used oil viscosity actually measuring value (mm²/s) .

Chapter 7 Rolling Bearing Mounting and Dismounting

The rolling bearing is an assembled part supporting the shaft running. The rings and rolling elements of the rolling bearing are manufactured through precise manufacturing processes. In order to preventing the surfaces from rusting, rust-proof oil or rust-proof grease has been put into the surfaces of rolling bearing rings and rolling elements after finishing the manufacturing process. Therefore, the rolling bearing should be washed clearly before mounting. If the rolling bearing is stored by the rust-proof oil, the gasoline or the kerosene can be used for washing. If the rolling bearing is stored by using the thick oil or grease for keeping rust-proof, the rolling bearing must be heating by using the light mineral oil, such as No. 10 machine oil or the grease dissolving and cooling, and then, washing by the gasoline or the kerosene. The bearing with seals or shields in double sides, it is not necessary to wash before mounting because grease has been put into the surfaces of rolling bearing rings and rolling elements after finishing the manufacturing. The bearings with grease for both lubrication and rust-proof are also not washed again while mounting. During the washing and mounting, it should ensure that the foreign particles can not be entered into the bearing because the little wear particle can cause the wearing for rings and rolling elements and increase the noise and vibration.

There are two ways for rolling bearing mounting and dismounting: first way is to press the tight fit ring into the shaft seat or to press onto the housing bore; second way is heating the bearing to expand the inner ring, or cooling the the bearing to contract the outer diameter. In these two ways, the bearing is easily mounted in the shaft seat or pushed onto the housing bore. When using the pressing method, the machinery force or the hydraulic force should be directly acted on the mounted or the dismounted ring through a special equipment. It is absolutely not permissible that the machinery

force or the hydraulic force is acted through the rolling elements, otherwise the dint or the pit would be existing on the surfaces of the raceway and the rolling elements. Thus the bearing vibration and noise would be increased, even some damage would be existing. When using the heating or the cooling method in bearing mounting and dismounting, the heating or cooling temperature should be seriously controlled. Because the rolling bearing working surface strength is influenced by the surface metallographic structure, if the temperature is too high, the tempering would be existing; if the temperature is too low, the metal surface would be brittle. Usually the highest temperature can not be over than 100°C , the lowest temperature can not be lower than -50°C . At the same time, it should be also considered that the heating should be uniform.

Practices have been proved that the incorrect installation is one of the reasons of the bearing earlier damage. It is very difficult to mount and to dismount the tight fit bearing ring. Therefore, while designing the support, it must be considered how to get the bearing mounting and dismount easier. For the large type rolling bearing mounting and dismounting of the heavy machine, it should be considered to design the special equipment, such as if using the high pressure oil mounting and dismounting device, the oil pipe and the pump should be designed. Here just some devices for the moderate type and small type bearing mounting and dismounting are introduced. The angular contact bearing mounting process is also the bearing clearance adjusting process, therefore while introducing the angular contact bearing mounting process, the bearing clearance adjusting process is also introduced.

7.1 Mounting and Dismounting of Cylindrical Bore Radial bearings

1. Mounting

Usually, for the non-separable bearings (the inner ring and outer ring can not be separable), the fit of the inner ring with the shaft is using the tight fit; the fit of the outer ring with the housing is using the loose fit. In

this case, the bearing can be firstly pressed on the shaft by pressure, and then the bearing together with the shaft are mounted into the housing bore. Figure 7-1 shows the mounting sleeves for the pressure installation. The sleeves are made by the soft metal, such as the copper or the soft steel. Figure 7-1a shows that the sleeve inner diameter is larger than the shaft diameter just for a little, the sleeve outer diameter is smaller than the inner ring rib for just a little. Figure 7-1b shows that the bearing is mounted onto the shaft and into the housing at the same time. In this case, a pressing plate should be added between the end faces of the sleeve and the bearing, its inner diameter is larger than the shaft diameter just for a little, its outer diameter is smaller than the bearing outer ring diameter for just a little. The pressing plate action is preventing the bearing outer ring from bending or blocking in the housing bore. The acting force can be the mechanical force, but also can be the hydraulic force. But it is absolutely not permissible to hammer directly to the ring end face. Because the ring belongs to the hardening thin wall component, it is easy to produce the crack by hammering.

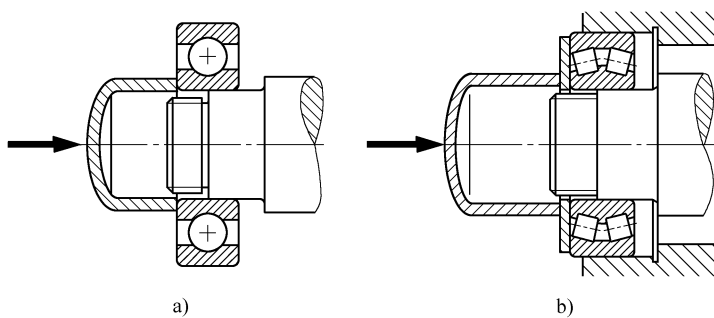


Figure 7-1 Mounting sleeve

a) mounting onto shaft b) mounting onto shaft and into housing bore at the same time

2. Heating mounting

The mounting force of interference fit ring is related to the bearing dimension and the interference amount. For the large type bearings, the commonly used method is the heating mounting method. Before mounting,

we put the mounted bearing into a clean oil bath, heating uniformly to $80 \sim 100^{\circ}\text{C}$, and then taking out from the oil bath and mounting onto the shaft. When heating, the bearing can not be put onto the oil bath bottom for preventing the depositing impurity from entering into the mounted bearing and it must be seriously monitored that the oil temperature can not be over than 100°C . The other electrical heating methods also can be used, such as the induction heating device, the electric furnace, or most simply, the heating plate.

3. Dismounting

When dismounting the non-separable bearing with the outer ring loose fit in the housing, the first step is to pull out the bearing together with the shaft, and then to dismount the bearing from the shaft. For the separable inner ring and outer ring, the first step is to dismount the shaft with the inner ring, and then to pull out the inner ring from the shaft; then to dismount the outer ring from the housing. Figure 7-2 shows this dismounting process. Figure 7-3 shows the process of dismounting the bearing from the shaft or the inner ring from the shaft by pressing machine, it can be also used to dismounting the bearing by the mechanical devices as shown in Figure 7-4 and Figure 7-5. While the bearing outer ring is the tight fit in the housing, the outer ring can be pressed by the pressing machine as shown in Figure 7-6.

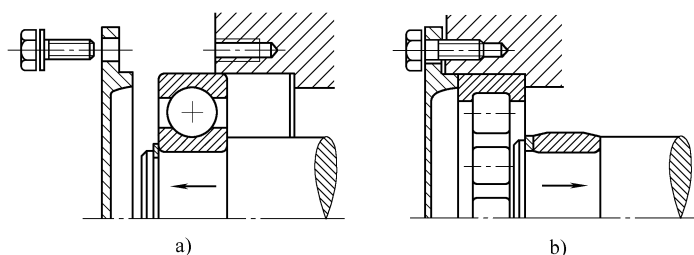


Figure 7-2 Dismounting bearing process

a) dismounting non-separable bearing b) dismounting separable bearing

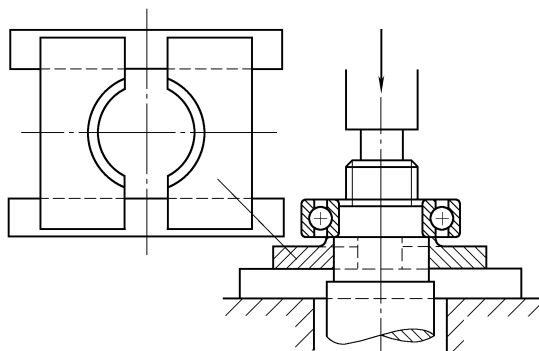


Figure 7-3 Pressing out shaft by pressing machine

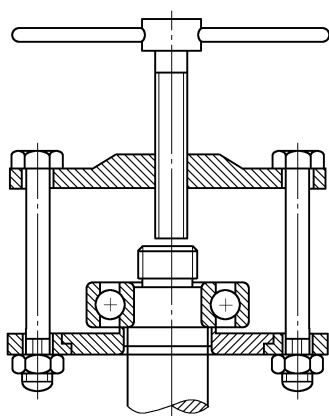


Figure 7-4 Pressing out shaft by threaded rod

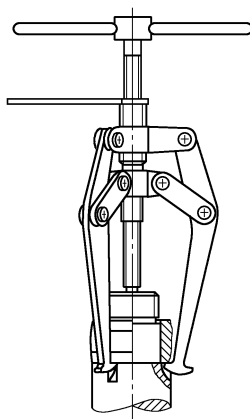


Figure 7-5 Pulling out inner ring threaded mechanism

4. Several convenient measures for mounting and dismounting

When designing the shaft abutment and the housing bore blocking rib, if designing a hole or a slot for placing the dismounting mechanism element on the shaft abutment and the housing bore blocking rib, the dismounting work would be very convenient. Figure 7-7a shows

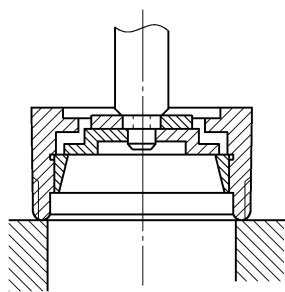


Figure 7-6 Pressing out outer ring by pressing machine

the machining slot on the shaft abutment for placing the dismounting mechanism element. Figure 7-7b shows the threaded hole on the housing bore blocking rib for placing the dismounting screw. Figure 7-7c shows the machining slot on the housing bore blocking rib for placing the dismounting element.

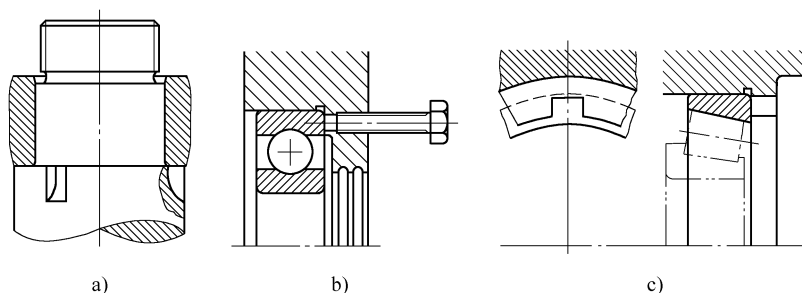


Figure 7-7 Several convenient measures for mounting and dismounting

7.2 Mounting and Dismounting of Tapered Bore Radial Bearings

1. Feature

The tapered bore bearing can be mounted onto the shaft with tapered seat, or mounted onto the tapered shape outer surface of the adapter sleeve or the withdrawal sleeve. These bearings are required to have the tight fit with the shaft. This tight fit is not achieved by the tolerances of the shaft diameter and the bore diameter, but is achieved by the axially pressing on the inner ring. When the inner ring is pressed onto the shaft seat with tapered shape, the inner ring is expanded, and thus the bearing radial clearance is reduced. Therefore, the reduction of the bearing radial clearance or the displacement on the taper can be used as whether a sufficient tight fit has been reached. While the taper degree is the standard taper degree 1:12, the bearing axial displacement amount on the taper is about 15 times of the radial clearance reduction. The interference fit amount 75% ~ 85% of the fitting surfaces can only be used for letting the inner ring expanding. Table 7-1 and Table 7-2 list the radial clearance reduction

and the relevant bearing axial displacement amount on the taper during installation of the cylindrical roller bearings with tapered bore and of the spherical roller bearing with tapered bore.

Table 7 – 1 Radial clearance reduction and the relevant bearing axial displacement amount on the taper during installation of the cylindrical roller bearings with tapered bore

bearing inner ring diameter/mm		requiring radial reduction/mm	axial displacement on taper 1: 12	
over	to		on shaft/mm	on sleeve/mm
40	50	0. 025 ~ 0. 030	0. 40 ~ 0. 50	0. 55 ~ 0. 60
50	65	0. 030 ~ 0. 035	0. 50 ~ 0. 55	0. 60 ~ 0. 70
65	80	0. 030 ~ 0. 040	0. 50 ~ 0. 65	0. 60 ~ 0. 75
80	100	0. 035 ~ 0. 045	0. 55 ~ 0. 70	0. 70 ~ 0. 85
100	120	0. 040 ~ 0. 050	0. 65 ~ 0. 80	0. 75 ~ 0. 90
120	140	0. 045 ~ 0. 055	0. 70 ~ 0. 85	0. 85 ~ 1. 00
140	160	0. 045 ~ 0. 060	0. 70 ~ 0. 95	0. 85 ~ 1. 05
160	180	0. 050 ~ 0. 065	0. 80 ~ 1. 00	0. 90 ~ 1. 15
180	200	0. 055 ~ 0. 070	0. 85 ~ 1. 10	1. 00 ~ 1. 20
200	225	0. 065 ~ 0. 080	1. 00 ~ 1. 25	1. 15 ~ 1. 35
225	250	0. 070 ~ 0. 085	1. 10 ~ 1. 30	1. 20 ~ 1. 45
250	280	0. 075 ~ 0. 095	1. 15 ~ 1. 45	1. 30 ~ 1. 60
280	315	0. 080 ~ 0. 100	1. 25 ~ 1. 55	1. 35 ~ 1. 65
315	355	0. 095 ~ 0. 115	1. 45 ~ 1. 75	1. 60 ~ 1. 90
355	400	0. 100 ~ 0. 125	1. 55 ~ 1. 90	1. 65 ~ 2. 05
400	450	0. 115 ~ 0. 140	1. 80 ~ 2. 20	1. 90 ~ 2. 30
450	500	0. 130 ~ 0. 160	2. 00 ~ 2. 50	2. 10 ~ 2. 60

(continued)

bearing inner ring diameter/mm		requiring radial reduction/mm	axial displacement on taper 1: 12	
over	to		on shaft/mm	on sleeve/mm
500	560	0. 140 ~ 0. 180	2. 20 ~ 2. 80	2. 30 ~ 2. 90
560	630	0. 150 ~ 0. 200	2. 40 ~ 3. 10	2. 50 ~ 3. 20
630	710	0. 180 ~ 0. 230	2. 80 ~ 3. 50	2. 90 ~ 3. 60
710	800	0. 210 ~ 0. 270	3. 20 ~ 4. 10	3. 30 ~ 4. 20
800	900	0. 230 ~ 0. 300	3. 60 ~ 4. 60	3. 70 ~ 4. 70
900	1000	0. 260 ~ 0. 340	4. 00 ~ 5. 20	4. 10 ~ 5. 30
1000	1120	0. 280 ~ 0. 370	4. 30 ~ 5. 60	4. 40 ~ 5. 70
1120	1250	0. 300 ~ 0. 400	4. 60 ~ 6. 10	4. 70 ~ 6. 20

Table 7 – 2 Radial clearance reduction and the relevant bearing axial displacement amount on the taper during installation of the spherical roller bearings with tapered bore

bearing inner ring diameter/mm		requiring radial reduction/mm	axial displacement on			
			taper 1: 12		taper 1: 30	
over	to		on shaft /mm	on sleeve /mm	on shaft /mm	on sleeve /mm
40	50	0. 025 ~ 0. 030	0. 40 ~ 0. 45	0. 45 ~ 0. 50		
50	65	0. 030 ~ 0. 040	0. 45 ~ 0. 60	0. 50 ~ 0. 70		
65	80	0. 040 ~ 0. 050	0. 60 ~ 0. 75	0. 70 ~ 0. 85		
80	100	0. 045 ~ 0. 060	0. 70 ~ 0. 90	0. 75 ~ 1. 00	1. 75 ~ 2. 25	1. 80 ~ 2. 40
100	120	0. 050 ~ 0. 070	0. 70 ~ 1. 10	0. 80 ~ 1. 20	1. 90 ~ 2. 70	2. 00 ~ 2. 80
120	140	0. 065 ~ 0. 090	1. 10 ~ 1. 40	1. 20 ~ 1. 50	2. 70 ~ 3. 50	2. 80 ~ 3. 60
140	160	0. 075 ~ 0. 100	1. 20 ~ 1. 60	1. 30 ~ 1. 70	3. 00 ~ 4. 00	3. 10 ~ 4. 20
160	180	0. 080 ~ 0. 110	1. 30 ~ 1. 70	1. 40 ~ 1. 90	3. 20 ~ 4. 20	3. 30 ~ 4. 60
180	200	0. 090 ~ 0. 130	1. 40 ~ 2. 00	1. 50 ~ 2. 20	3. 50 ~ 4. 50	3. 60 ~ 5. 00

(continued)

bearing inner ring diameter/mm		requiring radial reduction/mm	axial displacement on			
			taper 1: 12		taper 1: 30	
			on shaft /mm	on sleeve /mm	on shaft /mm	on sleeve /mm
over	to					
200	225	0. 100 ~ 0. 140	1. 60 ~ 2. 20	1. 70 ~ 2. 40	4. 00 ~ 5. 50	4. 20 ~ 5. 70
225	250	0. 110 ~ 0. 150	1. 70 ~ 2. 40	1. 80 ~ 2. 60	4. 20 ~ 6. 00	4. 60 ~ 6. 20
250	280	0. 120 ~ 0. 170	1. 90 ~ 2. 60	2. 00 ~ 2. 90	4. 70 ~ 6. 70	4. 80 ~ 6. 90
280	315	0. 130 ~ 0. 190	2. 00 ~ 3. 00	2. 20 ~ 3. 20	5. 00 ~ 7. 50	5. 20 ~ 7. 70
315	355	0. 150 ~ 0. 210	2. 40 ~ 3. 40	2. 60 ~ 3. 60	6. 00 ~ 8. 20	6. 20 ~ 8. 40
355	400	0. 170 ~ 0. 230	2. 60 ~ 3. 60	2. 90 ~ 3. 90	6. 50 ~ 9. 00	6. 80 ~ 9. 20
400	450	0. 200 ~ 0. 260	3. 10 ~ 4. 10	3. 40 ~ 4. 40	7. 70 ~ 10. 00	8. 00 ~ 10. 40
450	500	0. 210 ~ 0. 280	3. 30 ~ 4. 40	3. 60 ~ 4. 80	8. 20 ~ 11. 00	8. 40 ~ 11. 20
500	560	0. 240 ~ 0. 320	3. 70 ~ 5. 00	4. 10 ~ 5. 40	9. 20 ~ 12. 50	9. 60 ~ 12. 80
560	630	0. 260 ~ 0. 350	4. 00 ~ 5. 40	4. 40 ~ 5. 90	10. 00 ~ 13. 50	10. 40 ~ 14. 00
630	710	0. 300 ~ 0. 400	4. 60 ~ 6. 20	5. 10 ~ 6. 80	11. 50 ~ 15. 50	12. 00 ~ 16. 00
710	800	0. 340 ~ 0. 450	5. 30 ~ 7. 00	5. 80 ~ 7. 60	13. 30 ~ 17. 50	13. 60 ~ 18. 00
800	900	0. 370 ~ 0. 500	5. 70 ~ 7. 80	6. 30 ~ 8. 50	14. 30 ~ 19. 50	14. 80 ~ 20. 00
900	1000	0. 410 ~ 0. 550	6. 30 ~ 8. 50	7. 00 ~ 9. 40	15. 80 ~ 21. 00	16. 40 ~ 22. 00
1000	1120	0. 450 ~ 0. 600	6. 80 ~ 9. 00	7. 60 ~ 10. 2	17. 00 ~ 23. 00	18. 00 ~ 24. 00
1120	1250	0. 490 ~ 0. 650	7. 40 ~ 9. 80	8. 30 ~ 11. 0	18. 50 ~ 25. 00	19. 60 ~ 26. 00

While designing these kinds of rolling bearings, in order to leave the axial space required for the radial clearance reduction amount, the relevant axial displacement required for the radial clearance reduction amount should be considered. The radial clearance reduction amount is the difference between the clearance values prior to and after mounting. Therefore, during the pressing mounting, at first, the radial clearance prior to mounting must be measured, it must then be continuously monitored during mounting until

the required values and thus the necessary tight fit achieved, that is the original determining fitting is achieved. With the spherical roller bearings, it is important to measure the clearance at both roller sets simultaneously, because identical clearance values ensure that the inner ring is not laterally offset relative to the outer ring as shown in Figure 7-8.

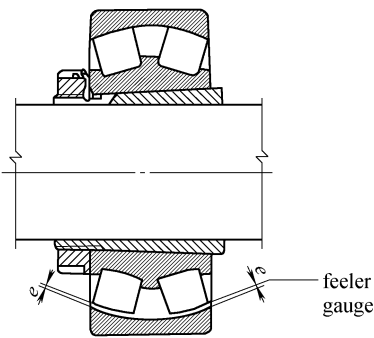


Figure 7-8 Spherical roller bearing radial clearance

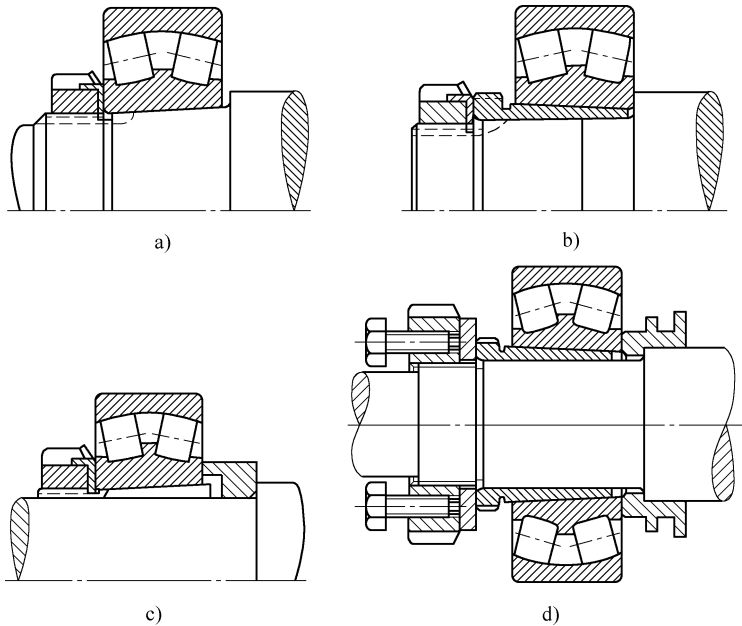


Figure 7-9 Tapered bore radial bearing mounting

2. Mounting and dismounting

Using the tapered seat of adapter sleeve and withdrawal sleeve to support the bearing with tapered bore, the mounting and dismounting of these kinds of bearings become more easy and convenient. But this method

can not be used for the heavy axial load applications.

The radial clearance reduction can be precisely controlled by means of a mounting nut using for mounting this kind of bearings. Figure 7-9 shows mounting a bearing with tapered bore by a mounting nut. Figure 7-9a shows mounting a bearing with tapered bore directly onto a shaft with tapered seat by a mounting nut. Figure 7-9b shows mounting a bearing with tapered bore onto a adapter sleeve by a mounting nut. Figure 7-9c shows mounting a bearing with tapered bore onto a withdrawal sleeve by a mounting nut. Figure 7-9d shows mounting a bearing with tapered bore onto a withdrawal sleeve by a mounting nut with screws.

Figure 7-10 shows dismounting a bearing with tapered bore.

Figure 7-10a shows dismounting a bearing with tapered bore by a top pressing nut to pull off the withdrawal. Figure 7-10b shows dismounting a bearing with tapered bore by a pressing nut with top pressing screw to add the pulling off force. Figure 7-10c shows dismounting a bearing with tapered bore by a dismounting sleeve. Figure 7-10d, and Figure 7-10e are the same means as Figure 7-10c, because the tapered surface can be self-locked, so after loosening the locking nut, it is also required to add a thrust force for separating the bearing with the shaft tapered seat. It should be considered to add force uniformly.

When mounting larger bearings, an annular piston press is commonly used as shown in Figure 7-11. Figure 7-11a shows pressing the bearing onto a tapered shaft seat. Figure 7-11b shows pulling out the withdrawal.

3. Heating mounting

Bearings with tapered bore can be mounted by the heating mean. Before mounting, the bearing working position should be determined in advance, and the axial position of the inner ring is controlled by an auxiliary ring, that is used exclusively for this purpose and its width u is obtained as the difference between the distance s of the bearing fitted snugly when cold and the amount of axial drive-up a required distance for a tight fit. The auxiliary ring will be moved out after cold. S is the distance between the bearing end face and the end cup before heating mounting, a is the axial displacement of the bearing for get the required interference fit amount, therefore $u = s - a$. The auxiliary ring width should be u .

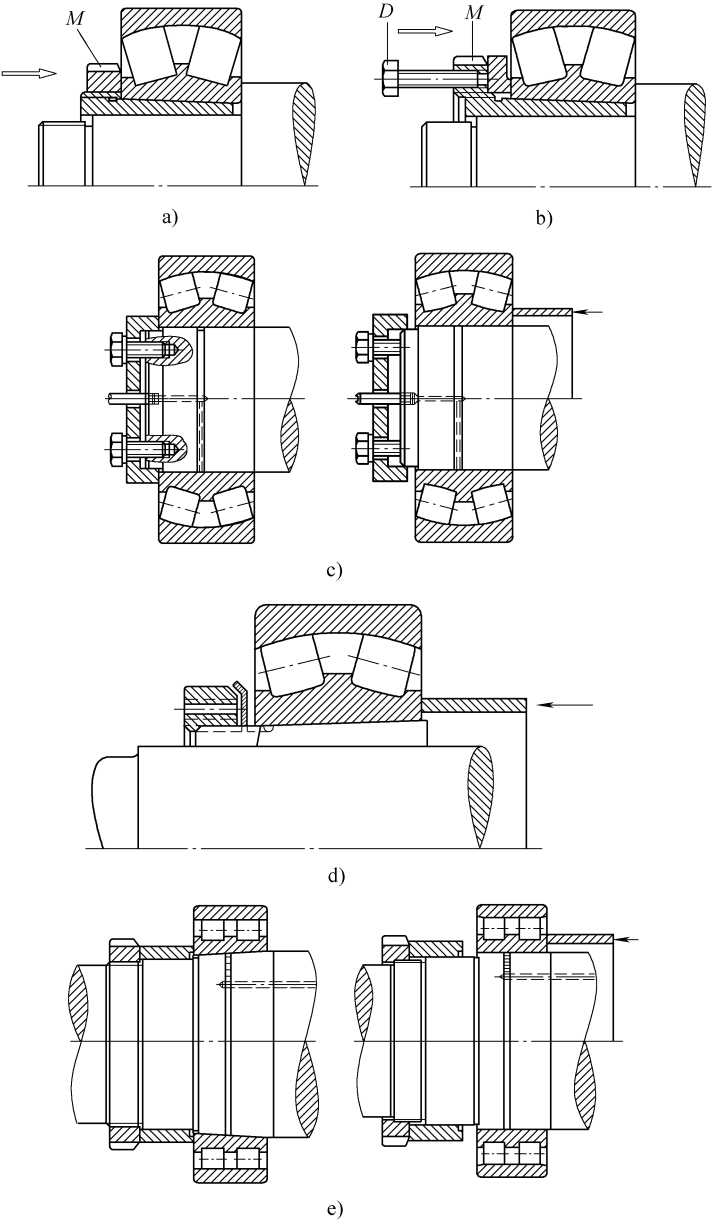


Figure 7 – 10 Dismounting radial bearing with tapered bore

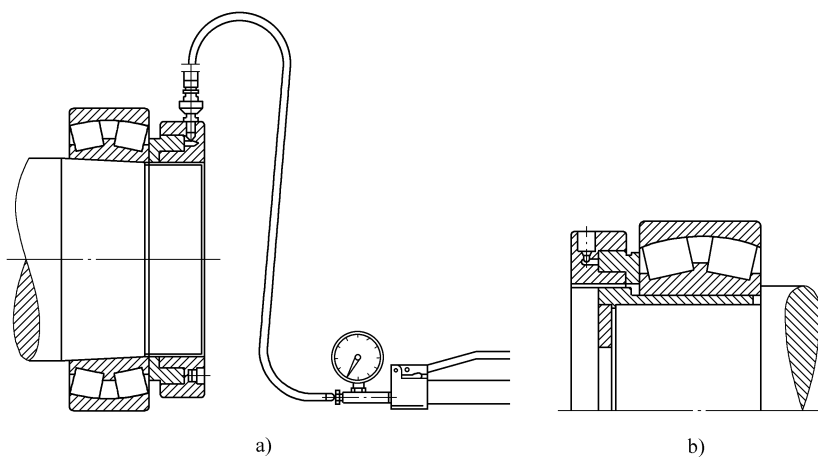


Figure 7-11 Annular piston press

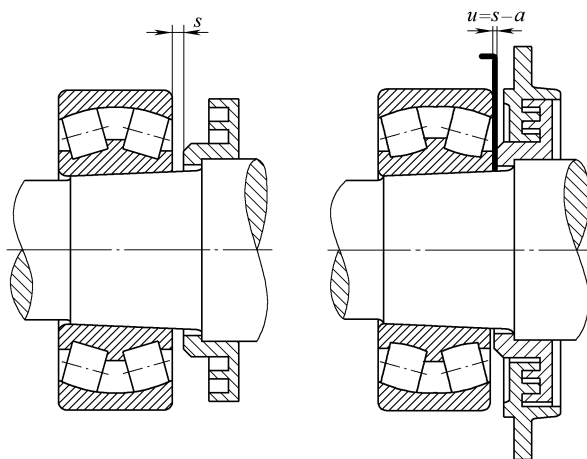


Figure 7-12 Heating mounting bearing with tapered bore

7.3 Mounting and Dismounting of Angular Contact Bearings

Angular contact ball bearing and tapered roller bearing both can only limit a single direction position moving, regardless of being a single

direction limiting support or being a double direction limiting support (fixed support). Usually, these bearings are paired mounted. When back-to back paired mounting, the one bearing outer ring wide end face is opposite to the other bearing outer ring wide end face. In this case, the bearing clearance reduction can be achieved by only pressing the inner ring moving to the outer ring. Because it is convenient to adjust the bearing clearance on the shaft and this support rigidity is large, there is no blocking problem. Therefore, this kind of support structure has been get comprehensively usage. Of course, in special case, pressing the outer ring onto the inner ring also can be used for diminishing the bearing clearance. Figure 7 - 13 shows the bearing clearance adjusting mean of back-to-back mounting. Figure 7 - 13a shows an example of a transporting car floating wheel support. Because of the large shocking force during working and the low speed, the clearance can be adjusted as a little. When adjusting, the shaft nut can be adjusted to such a degree that the wheel can be rotated smoothly by hand, and then rotating back to a fixed position to be fixed and locked by a crown screw with split pin. But it also can be fixed by a clocking nut and a washer with wings, or fixed by a stopping washer. Figure 7 - 13b shows an example of a car wheel hub support structure, that is resemble to the transporting car floating wheel support structure, but the difference is the higher speed. When adjusting, the shaft nut can be adjusted to such a degree that the wheel can be rotated smoothly by hand, and it is the proper stopping position of rotating half circle to one circle. At this time the bearing has been get a little pre-loading. When batch producing, the bearing clearance amount can be monitored by the dial gauge. Figure 7 - 14 shows a grinding machine grinding wheel support structure, the special features are the high rotational speed and the low load. And the high requirements for the rotating precision and the locating precision. Two single direction limiting position supports are used by two angular contact ball bearings. Two bearings in every support are tandem arrangement, and two single direction limiting position supports are back-to back arrangement mounting. In order to raise the support rigidity and the positioning precision, a thread spring is

placed between two bearing outer rings for pressing two inner rings, thus diminishing clearance and keeping pre-loading. This kind of support structure has been also get comprehensively usage.

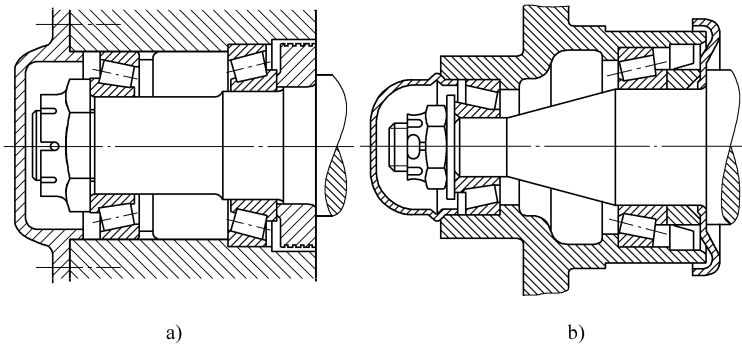


Figure 7-13 Clearance adjustment of back-to- back mounting

a) Transporting car floating wheel support b) Car wheel hub support structure

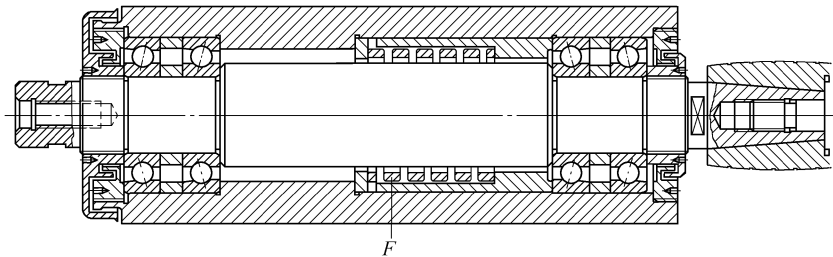


Figure 7-14 Grinding machine grinding wheel support structure

When using face-to-face arrangement mounting, two bearing outer ring narrow end faces are opposite. In this case, two inner rings can only be moved in the outer ring limiting range, the bearing clearance can only be changed by changing the outer ring position. Therefore, the bearing clearance adjusting is achieved by the outer ring axial moving, this also is a convenient method. The gap between the outer ring end face and the end cap can not be too small, usually is taking as 0.5 ~ 1mm. Figure 7-15 shows the clearance adjustment of face -to- face arrangement mounting, the thread sleeve, the cap washer are used respectively for adjusting the bearing

clearance. The adjusting ring can also be used.

In special condition, when two bearing outer ring axial positions can not be changed, an adjusting ring can be placed between the inner ring and the shaft abutment for adjusting the inner ring axial position.

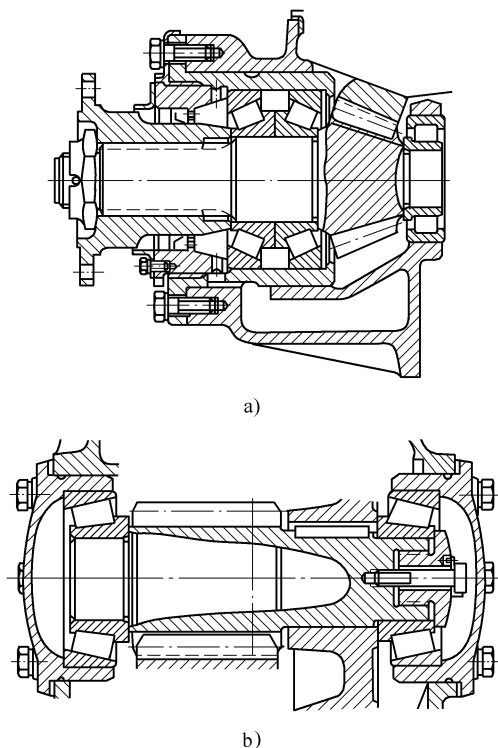


Figure 7-15 Bearing clearance adjustment of face-to face mounting

a) thread sleeve adjustment b) end cap washer adjustment

7.4 Mounting and Dismounting of Thrust Bearings

The single direction thrust bearing and the double direction thrust bearing both can not limit the radial position, but both only can limit the axial position. Therefore, these bearings must be used with other type bearings as a support. In order to prevent rolling element from harmfully

sliding on the raceway, when mounting thrust bearing, the axial clearance must be adjusted. There are some axial clearance adjustment methods, such as washer adjustment, bore using thread sleeve and spring adjustment. Figure 7-16 shows the above two adjustment methods.

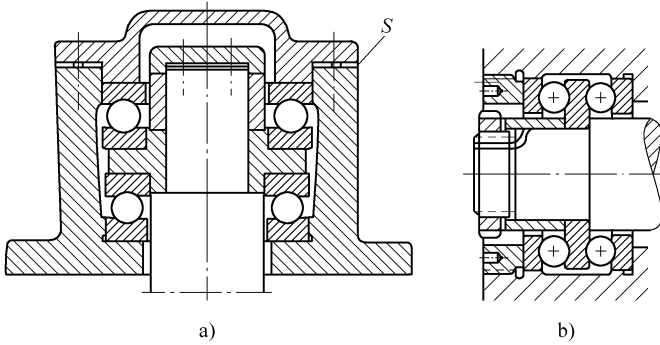


Figure 7-16 Thrust bearing clearance adjustment

a) washer adjustment b) bore using thread sleeve and spring adjustment

Figure 7-16 a, b both are double direction thrust bearings. Their shaft washers are rotating with the shaft, housing washers are stationary in the housing. For different washer thickness, the axial distance is different. While adjusting, two row ball sets should be adjusted to have pre-load. In this way, one ball set can be normally working in the raceway during unloading time. In order to prevent the rolling element from colliding the raceway in heavy machine large bearings, it is required to have pre-load between the roller set and the raceway. It is commonly used to place a spring for diminishing clearance and increasing pre-loading force.

7.5 Adjustment of Clearance of Combined Supports

Combined support is composed from two different type bearings. During designing the combined support, except for considering how to achieve radial positioning and axial positioning, it should be considered how to resolve clearance adjustment problem. Figure 7-17 shows a lathe machine spindle front support, that is composed from a double row cylindrical roller

bearing with tapered bore and two single row thrust ball bearings. The nut A is fixing the double row cylindrical roller bearing with tapered bore onto the tapered seat of the shaft for reducing the radial clearance. The nut B is adjusting the left thrust ball bearing clearance to zero. The spring is used to diminish the right thrust ball bearing clearance, and preloading two thrust ball bearings, for ensuring one ball set can be normally working in the raceway during unloading time.

The much little the support clearance is, the higher the rigidity and the working precision will be. But the working temperature raises with the clearance decrease. Therefore, during installing, the relation of the temperature with the clearance should be found, and the test running inevitably works. During test running process, after reasonable adjustment for the bearing clearance, the working temperature and the bearing clearance are nearly not changed in later working process.

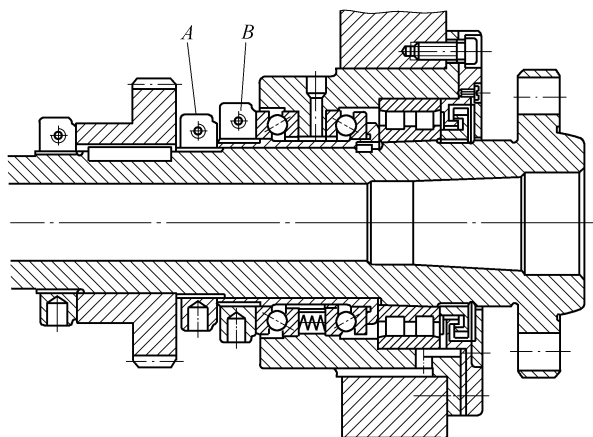


Figure 7-17 Combined support clearance adjustment

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